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A review on solar-powered closed physisorption cooling systems

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ABSTRACT

Cooling, refrigeration, and air conditioning processes are considered essential needs and major requirements for all human beings in our world today. However, the traditional vapor compression machines are dominating electricity consumers and their operation and propagation cause high electricity peak loads during the summer, especially in those countries with tropical climate. That is besides their refrigerants having high global warming as well as ozone layer depletion potentials. Providing cooling by utilizing a green energy such as solar energy is the key solution to electricity and pollution problems. Adsorption refrigeration systems that are driven by solar energy are mature technologies. They are proven to be suitable and applicable for refrigeration as well as air-conditioning applications. Solar adsorption cooling technology is divided into physisorption and chemisorption systems. The physisorption machines include open and closed cycle operation. This paper presents a review on previous researches and developments of the solar driven closed physisorption refrigeration systems. The discussion includes, experimental and numerical simulation studies as well as methods that are suggested to improve the system performance.

Contents

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1. Introduction

Energy is considered the heart and the continuous driving power for economic growth and the major requirement for technological development. The quality of life for people in a country is measured by the amount of energy they have. Increasing rate of population, economy, and the per capita energy consumption are the major forces that will be continue to cause increase in energy demand during the coming decades. We are certain that conventional energy types are neither reliable nor sustainable. Fossil fuels, such as oil, gas, and coal, are consumed at high rates and in some locations much more rapidly than they are produced. The world's reserves of oil are not large enough to be dependable in the future. Sustainable developments demand a sustainable supply of energy resources that, in the long run, are readily and sustainably available at reasonable cost and can be employed without causing adverse impacts. In addition to this, securing the supply of reliable and affordable energy and effecting a rapid transformation to a low-carbon, sustainable, efficient and environmentally benign systems of energy supply are central energy challenges facing us today. It is not an exaggeration to claim that the future of human prosperity depends on how successfully we tackle these issues. Renewable energy is that energy generated from natural resources and is renewable and naturally replenished, inexhaustible within the time horizon of humanity. Renewable types of energy are considered a perfect substitute for the traditional ones this include wind, geothermal, hydropower, biomass, wave, tidal and solar energy sources. Among these sources, solar energy comes at the top of the list due to its abundance and more equal distribution in nature than other types of renewable energy.

Solar energy is the electromagnetic radiation (including infrared, visible and ultraviolet light) released by thermonuclear reactions in the core of the sun. All the energy sources originate entirely from the sun with a few exceptions (e.g., nuclear, geothermal, and tidal energy). The surface of our planet receives a huge amount of energy flow from the sun. The solar radiation influx is expressed in terms of the solar constant S_c (i.e. the power transmitted by the sun's rays per unit cross sectional area at the mean distance of the earth). From measurements made on the earth and in spacecraft the mean value of the solar constant is considered to be equal 1368 W/m^2 [1]. Therefore, the total solar radiation transmitted to the earth's diametric plane area, which is estimated to 1.27516×10^{14} m^2 calculated at the earth mean radius which is 6371 km, is therefore approximately 1.74×10^{17} W [2].

Refrigeration and air conditioning processes mainly contribute in a considerable number of fields of human life. This includes food preservation, industrial process control, indoor air quality control, gas liquefaction, production of food and drink, and the chemical and pharmaceutical industry, and others. Along with the increasing trend in the worldwide economic growth, much more of the world's energy is being consumed to drive the conventional refrigerating and air conditioning appliances in both industry and buildings to meet the cooling demands. These traditional vapor compression machines are the dominating electricity consumers and their operation and propagation cause high electricity peak loads during the summer, especially in those countries with tropical climate. The energy consumption for air conditioning systems has recently been estimated to be 45% of the whole households and commercial buildings [3]. Furthermore, approximately 10–20% of all the electricity produced in the whole world is consumed by various kinds of the refrigeration and air-conditioning machines, as estimated by the International Institute of Refrigeration [4]. Moreover, the conventional vapor compression systems use non-natural working fluids and refrigerants like the chlorofluorocarbon (CFC), hydrochlorofluorocarbon (HCFC) or hydrofluorocarbons (HFC). These refrigerants have high global warming as well as ozone layer depletion potentials. Moreover, they contribute significantly in an opposite way to the international regulations.

Providing cooling by utilizing a clean and renewable energy such as solar energy is the key solution to decrease electricity requirements for cooling, peak electrical demand and energy costs without lowering the desired level of comfort conditions. For example, the Mediterranean countries may save 40–50% of their energy used for air conditioning by implementing solar driven air conditioning systems [5,6]. Even if 50% of present market of small AC systems can be replaced by solar powered systems, a considerable worth of electrical energy can be saved and a good amount of carbon credit can also be earned [3]. The development of solar refrigeration technologies became the worldwide focal point for concern because the peaks of requirements in cold coincide most of the time with the availability of the solar radiation. Solar refrigeration has the potential to improve the life quality for people who live in remote areas or areas with insufficient electricity. Moreover, the solar cooling technology can reduce the environmental impact raised by conventional air-conditioning systems. That is because the refrigerants used in these systems are environmentally benign, natural refrigerants and free from CFC. Therefore, these systems have zero ozone depleting as well as a zero global warming potentials.

Solar adsorption cooling technology is divided into physisorption and chemisorption systems. The physisorption machines include open and closed operating cycles. This paper presents a review on previous researches and developments of the solar driven closed physisorption refrigeration systems. We will be discussing various refrigeration technologies and advancements that were suggested to improve and enhance the performance of such technologies.

2. Principles of adsorption

Adsorption phenomenon has been known for long time and is increasingly used in many applications including separation, purification, and heat-powered green refrigeration technologies. The adsorption and absorption are mainly different processes. Adsorption is a surface phenomenon whereas absorption is a volumetric one. The heart of an adsorption process is usually a porous solid medium which provides a very large surface area and large pore volumes and therefore large adsorptive capacity [7]. The surface of the solid material is usually unsaturated and unbalanced. When surface is brought into contact with gas, there is an interaction between the unbalanced molecular forces at the surface and the gas molecular forces. That is because, solid surface tends to satisfy these residual forces by attracting and retaining on its surface the molecules, atoms, or ions of the gas. This results in a greater concentration of the gas or liquid in the near vicinity of the solid surface than in the bulk gas or vapor phase, despite the nature of the gas or vapor. The process by which this surface excess is caused is called adsorption [8]. The adsorption process may occur in two ways depending on the constraining force during the adsorption process. The first is the physical adsorption, or physisorption, and the second is the chemical adsorption, or chemisorption.

In the physisorption process, the adsorbate molecules are attracted to the adsorbent surface by the weak van der Waals force which are similar to the molecular forces of cohesion and are involved in the condensation of vapors into liquids. There are not any changes in the chemical composition of the adsorption pair. The enthalpy of adsorption is of the same order as the condensation heat of the gas. The binding molecules can be released by applying heat which usually does not exceed 80 kJ/mol. Physical adsorption is nonspecific and occurs between any adsorbate–adsorbent systems. Physical adsorbents can adsorb consecutive layers of the adsorbate gas and the thickness of the adsorbed phase is multimolecular.

The chemisorption process involves valency forces arising from sharing of electrons between the adsorbent and the adsorbate atoms. This results in a chemical reaction and forming a complex surface compound. The forces of these formed bonds are much stronger than the Van der Waals force in the physisorption. As a consequence, much more heat of adsorption is required to release the adsorbate, up to 800 kJ/mol. The adsorption heat of the sorbent is higher than the condensation heat of the refrigerant. The chemisorption process is specific and occurs between a certain gas and a certain corresponding adsorbent solid. The thickness of the adsorbed phase is unimolecular because only one layer of adsorbate reacts with the surface molecules.

The adsorption characteristics of a certain pairs depend on the nature of the adsorbate, the nature of the adsorbent, the reactivity of the surface, the surface area, and the temperature and pressure of adsorption. When a solid surface is exposed to a gas, the molecules of the gas strike the surface of the solid. Some of these striking molecules stick to the solid surface and become adsorbed, while some others rebound back. The rate of adsorption is large at the beginning because the whole surface is bare. It continues to decrease as more and more of the solid surface becomes covered by the adsorbate molecules. However, the rate of desorption increases because desorption takes place from the covered surface. The equilibrium is reached when the rate of adsorption is equal to the rate of desorption. At this point, the gas–solid system is said to be in adsorption dynamic equilibrium because the number of molecules sticking to the surface is equal to the number of molecules rebounding from the surface [8]. The amount adsorbed at the equilibrium for a given adsorbate–adsorbent system depends upon the pressure of the gas and the temperature of adsorption, the adsorption equilibrium can be represented as an adsorption isotherm at constant temperature, the adsorption bar at constant pressure, and the adsorption isoster for a constant equilibrium adsorption [8].

As in case of absorption, in both physisorption and chemisorption, the process of adsorption is exothermic and accompanied by the evolution of heat. Whereas, the process of desorption is endothermic and accompanied by absorption of heat. Therefore, this characteristic is used to produce the cooling effect in the refrigeration and air conditioning applications.

3. Adsorption cooling technologies

Adsorption refrigeration systems have some distinct advantages when compared with those absorption refrigeration system. These include, their simple control, the low operation and maintenances costs, the absence of vibration problems as a result of the absence of moving parts other than magnetic valves, and their very long lifetime. The lower driving temperatures of heat source that can be used and its wide range, make these systems more attractive. Adsorption systems can be powered by heat sources with temperature as low as 50 °C, whereas in absorption systems the source should be at least at 70 °C. Moreover, heat sources with temperature close to 500 °C can be used directly in adsorption without producing any kind of corrosion problem, while in absorption systems, severe corrosion would start to occur for temperatures above 200 °C [9]. Moreover, adsorption cooling machines have high storage capacity and energy density, and few problems associated to corrosion and crystallization. They are more suitable for working in hot and dry areas than absorption ones. That is because they do not require any extra cooling equipment for high ambient temperatures. Due to the fact that adsorption refrigeration systems use solid absorbent, they are suitable for conditions with serious vibration, such as in fishing boats and locomotives compared to the absorption systems. That is because the liquid absorbent in the absorption systems may flow from the generator to the evaporator or from the absorber to the condenser due to vibrations and consequently pollutes the refrigerant. Although the adsorption chiller systems have these advantages, their drawbacks are the requirements of special designs to maintain high vacuum, the large volume and weight relative to traditional refrigeration systems. That is besides the low specific cooling power SCP and the low coefficient of performance COP compared to absorption systems under the same conditions, the reason which limits the extensive application of this technology. Nevertheless, the use of this technology seems to be preferable for small cooling capacity chillers. Furthermore, these machines work intermittently where cooling production takes place during the night and desorption takes place during the day. Therefore, a cold storage, like ice or some cold mass, has to be provided for daytime cooling. However, two or more reactors or adsorbers can be used in order to provide continuous cooling production. That is besides the poor heat and mass transfer within the adsorbent, the adsorption deterioration of the adsorbent is also vital to the development and applications of the adsorption refrigeration technology [10]. However, enhancement of heat and mass transfer properties in the adsorbent bed, increasing the adsorption properties of the working pairs and a better heat management during the adsorption cycle lead to a more efficient system [11].

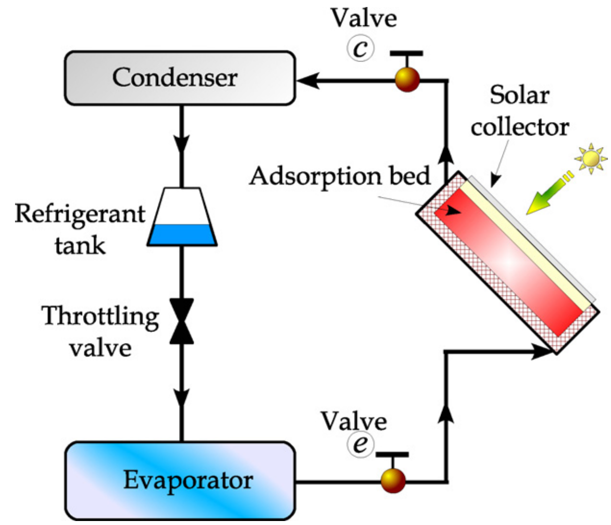

Fig. 1. Schematic diagram of the solar adsorption cooling system.

4. Working principle of the solar adsorption cooling system

The solar adsorption cooling system likes the basic vapor compression refrigeration machine except that the power compressor is replaced with a thermal or thermochemical compressor or a reactor. The main components of the basic system are shown in Fig. 1. This system consists of an adsorption reactor which is usually integrated with the solar collector, a condenser and an evaporator. The reactor is composed of a type of solid medium that has the affinity to physically or chemically adsorb and desorb the refrigerant vapor. This system operates intermittently through four different consecutive processes. These processes include: pressurization heating process, desorption at constant condenser pressure, depressurization cooling process, and adsorption at constant evaporator pressure [12].

At the beginning of the day and during the diurnal period, the reactor works as a generator or desorber. The reactor is isolated from both the condenser and the evaporator by valves *c* and *e* and is completely charged and saturated with the refrigerant. The pressure inside the reactor initially equals the evaporator pressure P_{evap}

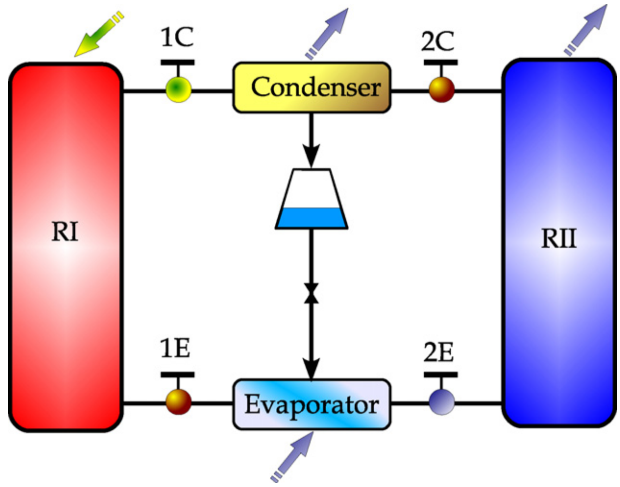

Fig. 2. Schematic diagram of the continuous two bed adsorption cooling system.

and its temperature is uniform and equals the ambient temperature T_{amb}. When the reactor is heated up by the incident solar radiation, both pressure and temperature inside the adsorbent are elevated. This heating process continues till the pressure reaches a value that equals the condenser pressure P_{cond}. This period is equivalent to the compression in the classical vapor compression refrigeration cycle. At the end of this process, valve *c* is opened and the adsorbate starts to desorb and flow toward the condenser. During this isobaric heating phase, the temperature continues to increase, and the refrigerant content inside the reactor continues to decrease as more adsorbate is being freed from the reactor. The desorbed refrigerant gas from the reactor condenses in the condenser at the diurnal outdoor temperature. When the adsorbate mass flow rate reaches its minimum value due to the decrease in the incident solar radiation, valve *c* is closed and the reactor starts the third process where the reactor is cooled down during the nocturnal period at constant volume and constant isoster till the pressure inside the reactor decreases to the evaporator pressure P_{evap}. The last process of the reactor cycle starts when valve *e* is opened and the refrigerant flows toward the reactor. This process consists of the refrigerant adsorption within the reactor and the production of cooling effect inside the evaporator. The adsorption process continues while the reactor is cooled at the constant evaporator pressure in order to remove the generated heat from the exothermic adsorption process.

The basic one bed cycle is an intermittent refrigeration cycle. The cooling effect is obtained only during one half of the cycle. Continuous and stable cooling effect with a constant flow of vapor from the evaporator can be achieved if two or more beds are used and operated out of phase. In this case, the generation and the cooling effect processes are shifted between the beds to guarantee that there is always a cooling effect all times during the cycle. The simplest continuous operation adsorption cycle consists of two beds that are operated out of phase. During the first half of the cooling cycle, Fig. 2, the first reactor RI is heated to start the regeneration desorption process. It is opened to the condenser and isolated from the evaporator, valve 1C is opened and valve 1E is closed, to allow cold production. Meanwhile the cooling effect takes place, the refrigerant vapor flow toward the second reactor RII, valve 2E is opened. This reactor in turns is isolated from the condenser, valve 2C is closed, and is cooled to allow the desorption of refrigerant vapor. In the second half cycle time, the heating and cooling steps are reversed. The cold production in the evaporator is due to the condensed vapor coming from the second reactor RII.

5. Adsorbents and working pairs

The performance of the adsorption cooling system depends mainly on the working pairs used. Since each pair differs from the other in terms of the physical and thermodynamic properties, the choice of the working pair has a great effect on the system performance and the technical concerns. The choice criteria depend on a number of important requirements. These include [13]:

- The refrigerant latent heat should be high, so the circulation rate of the refrigerant and absorbent can be minimized.
- The refrigerant/absorbent pair should not form a solid phase over the expected range of composition and temperature to which it will be subjected.
- The refrigerant should be much more volatile than the absorbent so the two can be separated easily without the need to a rectifier.
- The absorbent should have a strong affinity for the refrigerant under conditions in which absorption takes place. Strong affinity allows less absorbent to be circulated for the same refrigeration effect, reducing sensible heat losses, and allows a smaller liquid heat exchanger.
- Moderate operating pressure is required. High pressure requires heavy-walled equipment, and significant electrical power may be needed to pump fluids from the low-pressure side to the high-pressure side. Low pressure requires large-volume equipment and special means of reducing pressure drop in the refrigerant vapor paths.
- High chemical stability is required to avoid undesirable formation of gases, solids, or corrosive substances.

The most widely used working pairs which closely meet these requirements are activated carbon-methanol, activated carbon fibers–methanol, activated carbon-ethanol, activated carbon–ammonia, silica gel–water, and zeolite–water. The results obtained by Anyanwu and Ogueke [14] showed that zeolite/water is the best pair for air conditioning application while activated carbon–ammonia is preferred for ice making, deep freezing and food preservation. The maximum possible net solar COP was found to be 0.3, 0.19 and 0.16 for zeolite–water, activated carbon–ammonia and activated carbon-methanol, respectively, when a conventional flat plate solar collector is used.

5.1. Activated carbon granular and fiber adsorbent

Activated carbon is made by thermal decomposition of a carbonaceous material and is activated with carbon dioxide or steam at high temperatures, 700–1100 °C, to open the pores [15]. It is the most commonly used adsorbent in adsorption cooling systems due to its large interparticulate surface area, universal adsorption effect, high adsorption capacity, a high degree of surface reactivity and a favorable pore size. The most widely used activated carbons have a surface area of about 800–1500 m^2/g. Most of the adsorption on active carbons takes place in micropores (diam. < 2 nm) and only small amount in mesopores (diam. between 2 and 50 nm). The macropores (diam. > 50 nm) acting only as conduits for the passage of the adsorbate into the interior mesopores and the micropore surfaces [8]. The heat of adsorption of activated carbon pairs is lower than that of other types of physical adsorbent pairs [9]. Carbon fiber has a larger surface area, a better mass and heat transfer performance, and a more uniform pores than the granular one [16,17]. However, the higher contact thermal resistance between the fiber and the adsorber wall is its main disadvantage.

Methanol, ethanol, and ammonia as refrigerants are commonly used with both the granular and the fiber activated carbon. Methanol has low desorption temperature (about 100 °C), low adsorption heat (about 1800–2000 kJ/kg) and relatively high

evaporating latent heat and when used with active carbon, the pair attains large cyclic adsorption capacity. This makes activated carbon-methanol a suitable working pair to be used in solar driven cooling systems. Moreover, the active carbon based adsorption systems has the highest COP if methanol is used as a refrigerant [18]. The activated carbon fiber–methanol increases the COP by 10–20% and the adsorption capacity by 2–3 times [19]. However, the toxicity, the high vacuum required, and the low thermal conductivity of methanol besides its decomposition at temperatures higher than 120 are the main restriction factors on using methanol as a refrigerant [20]. Passos et al. [21] compared the performance of a solar adsorption refrigerator using methanol and three samples of commercial activated carbons. Rudenko et al. [22] studied the performance of solar refrigerating units using ammonia, methylamine, ethylamine, dimethylamine, methanol, and ethanol as refrigerants and activated carbon as adsorbents. Although the saturation pressure of ethanol is similar to that of methanol, ethanol latent heat of vaporization is 30% lower than that of methanol. Ammonia has a similar adsorption heat like methanol but its latent heat of vaporization and therefore the cooling capacity are higher compared with methanol. Moreover, activated carbon–ammonia systems have the merits of good heat and mass transfer performance due to the higher operation pressure of the system. Moreover, it is suitable for higher temperature operation. However, the adsorption capacity of ammonia system is lower than methanol. Furthermore, it is toxic and the system requires more care in sealing to prevent leakage due to the high pressure.

5.2. Silica gel adsorbent

Silica gel is a partially dehydrated form of polymeric colloidal silicic acid with a chemical composition that can be expressed as $(SiO_2)nH_2O$ [15]. Water as a refrigerant is commonly used with silica gel adsorbent. The adsorption–desorption action of silica gel for water vapor is a purely physical and can be expressed by the following equation:

$$SiO_2 \cdot (n-1)H_2O(\text{solid}) + H_2O(\text{vapor}) \leftrightharpoons SiO_2 \cdot nH_2O(\text{solid}) + \Delta H_{ads}$$

where ΔH_{ads} is the adsorption heat. When the particles become saturated, they do not suffer any change in size or shape, and even when completely saturated the particles seem to be perfectly dry [23].

The most attractive feature of silica gel–water pair is its ability to work at low temperatures, making it suitable for solar cooling applications. This pair can be driven with a heat source of temperature of 50 °C [24]. Furthermore, water is an ideal refrigerant which is non-toxic and has much larger latent heat than that of methanol or other traditional refrigerant. However, the limitations on this pair are the low adsorption capacity and the low vapor pressure of the water that limits the mass transfer. That is besides, the impossibility to have temperatures less than 0 °C with the minimum evaporating temperature is a few degrees Celsius. Moreover, the desorption heat is relatively high, 2500 kJ/kg and the desorption temperature is limited to maximum value of 120 °C [9]. Therefore, the applications are limited to air conditioning and chilling.

5.3. Zeolite adsorbent

For zeolite–water working pairs, the adsorption and desorption heat is higher than other pairs, about 3300–4200 kJ/kg. Their adsorption and desorption temperatures are high, about 70 and 200 °C, respectively. Therefore, this pair is employed in cooling systems that have heat source between 200 and 300 °C. As a consequence, the performance of zeolite/water pair is low for low temperature heat sources compared with that of activated carbon-methanol systems. As in the silica gel–water systems, it is impossible to produce evaporation temperatures below 0 °C besides the bad mass transfer performance due to the low working pressure.

6. Solar physisorption cooling systems

There are various types solar driven cooling systems that work based on the adsorption technology. These types are categorized, depending on the nature of the adsorption phenomenon, to physisorption and thermochemical adsorption systems. The thermochemical adsorption is just a type of the chemisorption in which the process is activated thermally.

Physical adsorption refrigeration has been studied extensively with different systems and operating thermodynamic cycles. Many theoretical and experimental studies have been introduced including new cycles, new system designs, new methods, and new working pairs. The physisorption cooling systems are subclassified into open and closed systems. In the following sections, the solar physisorption cooling technologies that are based on a closed cycle are being explained and discussed. These include: the basic closed cycle system, the freeze cooling tube system, and other advanced systems.

7. The basic single bed physisorption cooling cycle

The basic physisorption refrigeration thermodynamic cycle is shown on the conceptual Clapeyron diagram, Fig. 3. Since refrigerant phase change is monovariant, its equilibrium state is defined only by one variable, either by the pressure or the temperature. The dynamic equilibrium condition of the refrigerant is represented by the line on the left in Fig. 3. The condition of the gas–solid equilibrium in the reactor is bivariant and requires two variables to be specified, the concentration ratio and either the temperature or the pressure. Therefore, the temperature–pressure dynamic equilibrium of the gas–solid reactor is represented in Clapeyron diagram by the line in the middle for high concentration and the line in the right for the lower concentration ratios. The working region of the adsorption reactor spans from the lower concentration adsorption equilibrium line to the higher concentration adsorption equilibrium line. Whereas the working region of the refrigerant spans from the lower concentration adsorption equilibrium line to the refrigerant vapor–liquid dynamic equilibrium curve.

The adsorbent–adsorbate reactor cycle consists mainly of four processes: pressurization process at a constant volume and concentration (isosteric heating process 1–2), desorption at constant pressure (isobaric heating process 2–3), depressurization at constant volume and concentration (isosteric cooling process 3–4), and adsorption at constant pressure (isobaric cooling process 4–1) [12]. At the beginning of the day, state (1), the reactor is isolated from both the condenser and the evaporator and is completely charged and saturated with the refrigerant. The pressure inside the reactor initially equals the evaporator pressure P_{evap} and its temperature is uniform and equals the ambient temperature T_{amb}. When the adsorbent starts to heat up, along with the refrigerant, by the incident solar radiation, both pressure and temperature inside the adsorbent are elevated. This constant concentration heating phase continues till point (2) where the pressure reaches a value that equals the condenser pressure P_{cond}. This period is equivalent to the compression in the classical vapor compression refrigeration cycle. At state (2), the adsorbate starts to desorb and flow toward the condenser and then toward the expansion device where it is self-cooled before it enters the evaporator. During this isobaric heating process, the temperature continues to increase, and the adsorbate concentration continues to decrease as more adsorbate is being freed from

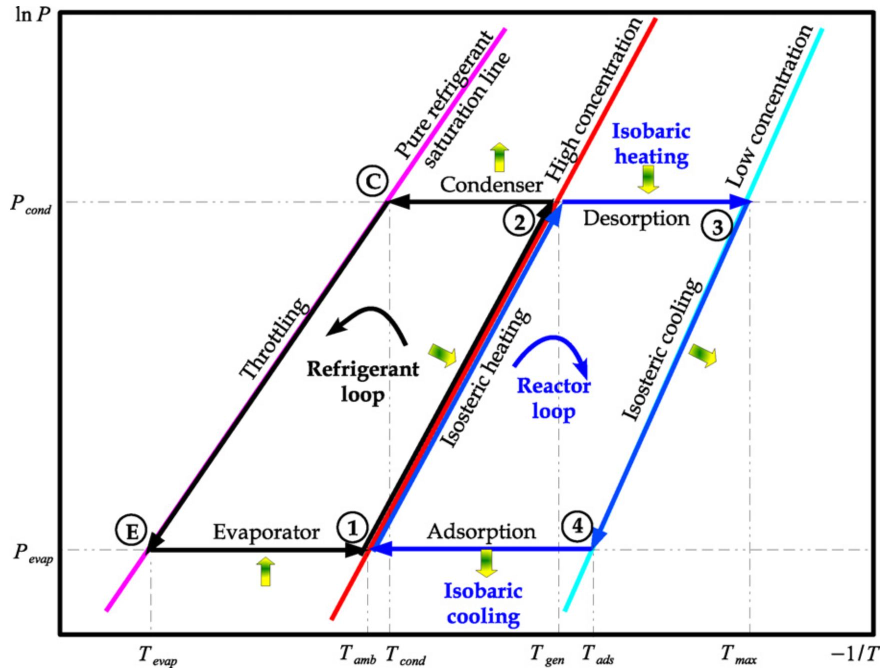

Fig. 3. Clapeyron diagram for the basic single bed physisorption thermodynamic cycle.

the reactor. When the adsorbate temperature reaches the maximum allowed value T_{max} at state (3), the reactor is isolated from the condenser and it starts the third process. It is cooled down at constant volume and constant isoster till the pressure inside the reactor decreases to the evaporator pressure P_{evap}, point (4). The last process of the reactor cycle starts at the night, point (4), when the refrigerant flows toward the reactor. The adsorption process continues while the reactor is cooled at the constant evaporator pressure till the higher cycle isoster at point (1).

It should be stated that the single stage basic adsorption heat pump cycle can be completely identified by the ambient or condensation temperature $T_{amb} \approx T_{cond}$ and the required cold production temperature T_{evap}. By referring to the basic Clapeyron diagram, Fig. 3, the evaporator pressure P_{evap} is equal to the refrigerant saturation pressure corresponding to T_{evap}. The condenser pressure P_{cond} also equals the refrigerant saturation pressure corresponding to T_{amb}. State (1) is specified by T_{amb} and P_{evap}. State (2) and therefore the generation temperature T_{gen} are specified by P_{cond} and the isoster line passing through state (1). State (3) and consequently the low concentration line are limited by the heat source available temperature T_{max}. State (4) is bounded by the evaporator pressure and the low concentration line. It is noted that the cycle will not be operational for a given T_{amb} and T_{evap} if the heat source temperature T_{max} is less than the generation temperature T_{gen}. The evaporating temperature lift is the difference between condensing temperature and evaporating temperature $\Delta T_{evap} = T_{cond} - T_{evap}$. The regeneration temperature lift is the difference between the generation temperature and the condenser temperature $\Delta T_{reg} = T_{gen} - T_{cond}$.

The single stage basic cycle is suitable for solar as well as for waste heat recovery refrigeration applications. Although it is simple and easy to be constructed, its low COP is the major problem. Many researches and studies have been done both theoretically and experimentally. The flat plate solar collector has been widely used in solar powered adsorption cooling systems. In this case, the adsorption bed is usually integrated inside the collector. A flat-plate solid-adsorption refrigeration ice maker has been built for demonstration purposes using activated carbon/methanol pair [25]. Li et al. [25] constructed an adsorbent bed of two flat-plate collectors, with a total surface area of 1.5 m^2. The adsorption solar refrigerator designed and constructed by Anyanwu and Ezekwe [26] has a flat plate type collector/generator/adsorber of effective exposed area of 1.2 m^2. The experimental results for a silica gel/water tubular reactor integrated with 2 m^2 double glazed flat plate collector give a gross solar cooling COP of 0.19 [27]. An adsorptive solar refrigerator was built and tested by Buchter et al. [28] with activated carbon/methanol pair a 2 m^2 single glazed solar collector. During the test period, irradiance was between 19 and 25 MJ m^{-2} and the ambient temperature was relatively warm with an averagely 27.4 °C at sunrise and 37.4 °C at mid-afternoon. The experimental values of the gross solar COP were found to be between 0.09 and 0.13. Metallic solar collectors with fins have been used to increase the thermal conductivity in solar collectors. However, this approach has a negative effect due to solar energy loss by reflection and heat loss resulting from the sensible heat of the metal. For this reason, a direct-radiation absorption collector was proposed by Tangkengsirisin et al. [29].

Collector types other than the conventional flat plate have been used with the solar adsorption heat pumps. Headley et al. [30] used a CPC solar collector of concentration ratio 3.9 and aperture area 2 m^2 to power an intermittent ice maker using activated charcoal/methanol pair. Up to 1 kg of ice at an evaporator temperature of −6 °C was produced, with the net solar COP being of the order of 0.02. Maximum receiver/adsorbent temperature recorded was 154 °C on a day when the insolation was 26.8 MJ m^{-2}. The major advantage of this system is its ability to produce ice even on overcast days (insolation about 10 MJ m^{-2}). A compound parabolic concentrator is used by González and Rodríguez [31,32] in their experimentation for the adsorption cooler that attained a measured solar COP ranged from 0.078 to 0.096. Niemann et al. [33] designed and constructed a collector consisting of an evacuated tubular collectors with external parabolic circle concentrators PCC for process heat generation up to 150 °C. The thermal energy is used to operate an ammonia–carbon ice maker generating block-ice for the fishing industry in developing countries.

The performance of the solar refrigerator is studied experimentally by many authors. The experimental results by Li et al. [25] showed that their machine can produce 4–5 kg of ice after receiving 14–16 MJ of radiation energy with a surface area of 0.75 m^2,

while producing 7–10 kg of ice after receiving 28–30 MJ of radiation energy with 1.5 m^2. In [26], activated carbon-methanol is used as the adsorbent–adsorbate pair. Adsorbent cooling during the adsorption process is both by natural convection of air over the collector plate and tubes and night sky radiation facilitated by removing the collector box end cover plates. Ambient temperatures during the adsorbate generation and adsorption process varied over 18.5–34 °C. The refrigerator yielded evaporator temperatures ranging over 1.0–8.5 °C from water initially in the temperature range 24–28 °C. Accordingly, the maximum daily useful cooling produced was 266.8 kJ/m^2 of collector area. An adsorption ice maker has been used for carrying out experiments under real solar radiation intensity conditions [34]. Different performance variations of the solar ice maker are obtained under the effect of sky cloudy cover. The experimental results indicated that the performance of the solar ice maker was severely affected by the sky cloudy cover, and no ice would be obtained if cloudy conditions prevailed for intervals exceeding 3 h. Using the activated carbon AC35-methanol pair, Lemmini and Errougani [35,36] built and tested a SAR system. Experimental results show that the unit produces cold even in rainy and cloudy days and the temperatures achieved by the unit can be less than −11 °C for days with a very high irradiation. The solar coefficient of performance (cooling energy/solar energy) ranges between 0.05 and 0.08 for an irradiation between 12 and 28 MJ/m^2 and a daily mean ambient temperature around 20 °C. Luo et al. [37] performed a year round performance tests of the solar ice maker. Test results show that the COP (coefficient of performance) of the solar ice maker is about 0.083–0.127, and its daily ice production varies within the range of 3.2–6.5 kg/m^2 under the climatic conditions of daily solar radiation on the surface of the adsorbent bed being about 15–23 MJ/m^2 and the daily average ambient temperature being within 7.7–21.1 °C.

The theoretical and simulation work has been extensively presented in the literature. Most of these studies were aiming at evaluating the performance and studding its dependency on other system parameters. A parametric study of the effects of collector and environmental parameters on the performance of solar refrigerator is analyzed by Li and Wang [38]. The parametric effects study included the effects of heat transfer fins, contact thermal resistance, adsorbent thermal conductivity, packing density, number of glazing, and selective coating. The environmental effects study included the effects of the solar radiation intensity, condensing temperature, and evaporating temperature. A thermodynamic optimization of a solar driven adsorption refrigeration system was conducted by Alam et al. [39] to analyze the optimum conditions for which the maximum refrigeration effect can be achieved. The description and operation of a solar-powered ice-maker with the solid adsorption pair of activated carbon and methanol was presented by Sumathy and Zhongfu [40]. A domestic type of charcoal is chosen as the adsorbent, and a simple flat-plate collector with an exposed area of 0.92 m^2 is employed to produce ice of about 4–5 kg/day. The system could achieve solar refrigeration COP of about 0.1–0.12. In a similar study, Leite and Daguene [41] obtained an average net solar COP of 0.13 corresponding to 7–10 kg/day of ice production per square meter of solar collection surface, respectively, with solar irradiations ranging from 20 to 23 MJ/m^2. Anyanwu et al. [42] performed a transient analysis and performance prediction of a single bed adsorption solar refrigerator, using activated carbon/methanol pair and integrated within a tubular solar flat plate collector. A study of the effects of different collector design parameters on the performances of a solar powered solid adsorption refrigerator is presented [43]. The refrigerator uses activated carbon/methanol as the adsorbent/refrigerant pair. The parameters tested are the collector plate emissivity/absorptivity combination, adsorbent packing density, tube spacing, outer tube outside diameter, adsorbent thermal conductivity, heat transfer coefficient at adsorbent/tube interface, and adsorbent tube/collector plate materials combination. Improvements in the ranges of 29–38% for COP and 26–35% for condensate yield were obtained with optimal choices of tube spacing, adsorbent packing density and collector plate/adsorbent tube material combinations.

El Fadar et al. [44] presented a simulation study of solar adsorption cooling machine in which the reactor is heated by a parabolic trough collector and is coupled with a heat pipe. This reactor contains the activated carbon and ammonia pair. In comparison with other systems powered by flat plate or evacuated tube collectors, the predicted results, have illustrated the ability of the proposed system to achieve a high performance due to high efficiency of PTC, and high flux density of heat pipe. Tashtoush et al. [45] used a multidimensional curve-fitting procedure to fit experimental as well as theoretical data relating the COP value to one of the three temperature variables; condensation, evaporation, or generation temperatures. A cylindrical adsorber heated by solar energy contains an activated carbon–ammonia pair and is composed by many cylindrical tubes welded using external fins is studied theoretically be Louajari et al. [46]. Using real solar irradiance data they showed that the performances of the solar adsorption refrigerating machine with an adsorber equipped with fins are higher than the machine without fins.

Mathematical models describing the physical operation of the system have been widely introduced. Anyanwu et al. [47] presented a transient simulation model of a solar adsorption refrigerator using activated carbon/methanol pair. The peak plate and tube surface temperatures and methanol generation are predicted to be within 3%, 2.5% and 4%, respectively. Based on the observed agreement between measured and predicted values, parametric study of the refrigerator system was undertaken in order to optimize the design. Li and Wang [48] presented a two-dimensional uniform pressure transient model to describe the heat and mass transfer in an adsorbent bed for a flat plate solar ice maker. The model is validated by experimentations and is used to analyze and predict the performance of an intermittent solar powered solid refrigerator.

8. The solar powered adsorption cooling tube system

The adsorption cooling tube ACT is a very interesting technology to produce cold from solar or waste low-grade energy. This technology has drawn increasing attention because of its advantages of no moving parts, compact structure and use of low grade heat as the heat source [49]. Besides, this special type of cooling systems has a compact structure because the adsorbent bed, condenser, and evaporator, are all integrated in one tube. That is besides the simplicity of the system because it does not require any valve control operation. The first solar-powered adsorption cooling tube was introduced in 1998 by Zhenyan et al. [50]. This solar ACT consists of a glass tube that is closed at both ends and concludes the collector/generator at the upper part and the condenser and the evaporator at the lower part. Fig. 4(a) shows the construction of the solar ACT system. The adsorption bed has a central pours tube representing a mass transfer channel for the refrigerant vapor. The thermodynamic principle of operation of the ACT cycle is not the same as the basic cycle however both cycles are intermittent in operation. Fig. 4(b) describes the ACT thermodynamic cycle on Clapeyron diagram. A complete cycle of this non-valve controlled system constitutes two processes or two half-cycle. The first half cycle is the heating–desorption–condensation process, path 1–2. This process takes place during the daytime where the generator is heated up by solar radiation and the refrigerant vapor starts to desorb from the bed. The desorbed refrigerant vapor is condensed in the condenser part of the solar ACT and by gravity; it is collected in the lower evaporator end of the outer glass tube.

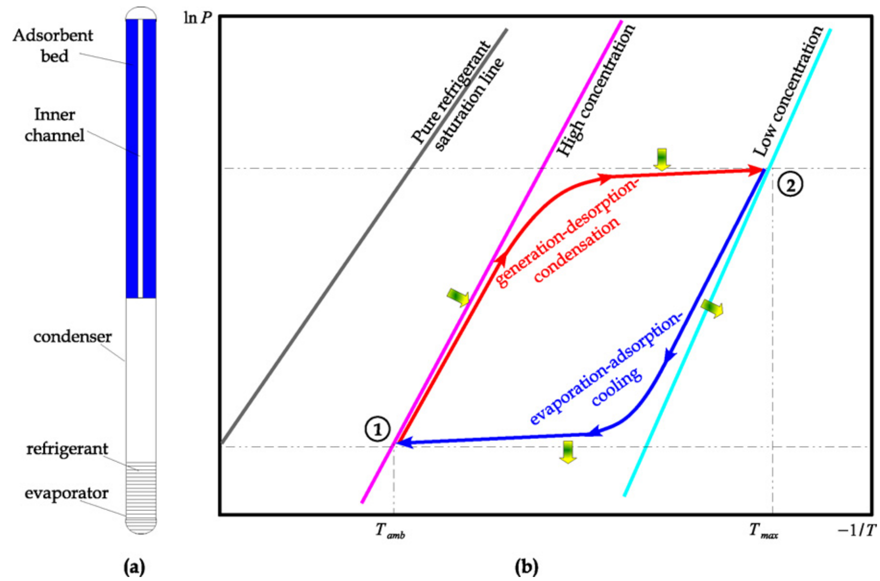

Fig. 4. The ACT system: (a) schematic diagram and (b) Clapeyron diagram [49].

During this process, both the bed temperature and the system pressure are continuously increasing whereas the concentration is decreasing. The second half cycle takes place during the nighttime where the evaporation–adsorption–cooling process, path 2–1, occurs. The condensed refrigerant in the lower end of the outer glass tube begins to evaporate, producing the cooling effect, and be re-adsorbed by the adsorbent bed which is cooled during this period by natural convection. In this process, both the bed temperature and the system pressure are continuously decreasing whereas the concentration is increasing, Fig. 4(b).

Zhenyan et al. [50] introduced an experimental study of the solar ACT using a compound zeolite-active carbon adsorbent (CZACA) which has high absorptivity to solar radiation and thus it can absorb solar energy directly with a dramatic increase in the heat transfer rate to adsorbent bed. Water is used as the refrigerant and a parabolic concentrating solar collector is used to heat up the generator. The solar-powered adsorption refrigeration module by Khattab [51] consists of a modified glass tube having a circular generator (sorption bed) at one end and a combined evaporator and condenser at the other end. Activated carbon-methanol pair is used and the charcoal is mixed with small pieces of blackened steel to enhance the heat transfer in the sorption bed. As a result of using the optimum ratio of the steel additives in the bed, the yearly average ice production increases from 0.23 to 0.25 kg/day, the yearly average bed efficiency increases from 55.2% to 58.5%, and the yearly average net COP increases from 0.146 to 0.1558.

Wang and Zhang [52] designed an adsorption cooling tube and an adsorption heat pump with multi-cooling tubes and use silica gel–water pair, Fig. 5. The adsorber, the condenser and the evaporator are all housed in the shell composed of one tube and two hemispherical heads. The adsorber in the upside of the tube consists of silica gel, fins, metal mesh, spring and shell. The fins are annular, and their center is the mass transfer channel surrounded by metal mesh and spring. The bottom of the cooling tube is the evaporator whose evaporating surface is coated with a porous layer to enhance heat transfer. The condenser in the middle of the cooling tube is a plain tube. In order to decrease the heat transferred from the hot part to the cold part, one thermal insulation section with thin wall and long length tube are reserved between the adsorber and the condenser as well as between the condenser and the evaporator. The process pipe in the extreme of the cooling tube will be isolated from the atmosphere after the cooling tube is evacuated and the refrigerant is charged into it. Fins can be installed out of the cooling tube or not, depending on the cooling/heat media. In this work, no fins out of the tube are installed because of the cooling/heating media. Their simulation results showed that the coefficient of performance and specific cooling power reach about 0.5 and 85 W/kg adsorbent, respectively, at the hot water temperature of 85 °C.

As discussed by Ma et al. [53], the first configuration of the solar ACT system has a solar COP value of about 0.094. This low performance is mainly due to the considerable heat loss from the outer glass tube. Moreover, the sensible heat and adsorption heat of the adsorbent bed are not taken out during the adsorption process. Some adjustments and modifications were made to the first configuration of the solar driven ACT to solve these existing problems and to improve its performance. Ma et al. [54] presented a new configuration of the solar ACT, Fig. 6. This configuration has an outer and inner glass tubes jointed together at the top end and are both closed at the lower ends. The outer glass tube is applied to reduce the heat loss and enhance the efficiency of the solar collector. The inner glass tube is inserted into the absorbent bed and used to cool the adsorbent bed during the adsorption process and therefore, sensible heat and adsorption heat of the adsorbent bed can be taken out during the adsorption process. The condenser is located at the top end of the tube whereas the evaporator is located at the lower end. To enhance the solar collector efficiency and reduce the heat loss from the system, the adsorption bed is confined with a third glass tube which is evacuated and represents the solar collector. Three vapor passages are arranged symmetrically in the adsorbent bed for a better mass transfer of the refrigerant vapor into and from the adsorbent bed. This configuration allows a better heat insulation of the solar collector and higher efficiency of heating due to using the outer glass tube. Furthermore, the condenser and evaporator are separated and located at two ends, which helps to avoid heat losses. The desorbed refrigerant vapor flows upward in the vapor passages and is condensed in the upper end of the outer glass tube during the heating–desorption–condensation process. At the second process, evaporation–adsorption–cooling, the condensed refrigerant flows into the lower end of the outer glass tube to evaporate and be re-adsorbed by the adsorbent bed. Ma et al. [54] presented experimental work to evaluate the performance of a solar-powered adsorption cooling tube using the working pair of zeolite 13X–water. The experimental results show that the solar adsorption cooling tube is capable of producing a

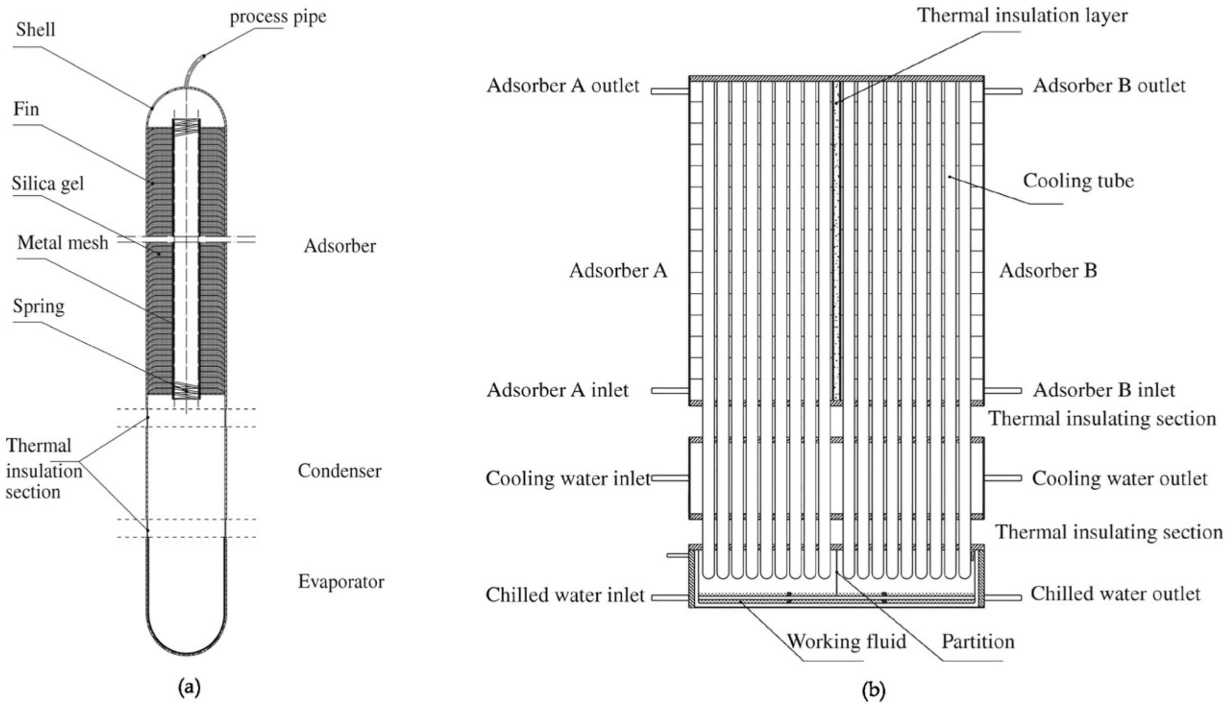

Fig. 5. The ACT by Wang and Zhang [52]: (a) structure of the cooling tube and (b) AHP composed of multi-cooling tubes.

refrigeration capacity of about 276 kJ with a solar COP of 0.22. In another study, a the same cooling tube configuration that uses zeolite 13X–water as a working pair is discussed experimentally by Ma et al. [53]. The experimental investigations of three generations of the solar-powered adsorption cooling tube were carried out to evaluate the system COP. The experimental results show that the COP values were 0.094, 0.161, and 0.182 for the first, second, and third generations of the solar-powered adsorption cooling tube, respectively. The freeze proof solar cooling tube, which can produce cooling capacity with the refrigerant temperature below 0 °C using active carbon-methanol as working pair, was firstly designed and introduced by Zhao et al. [55]. Their experimental results concluded

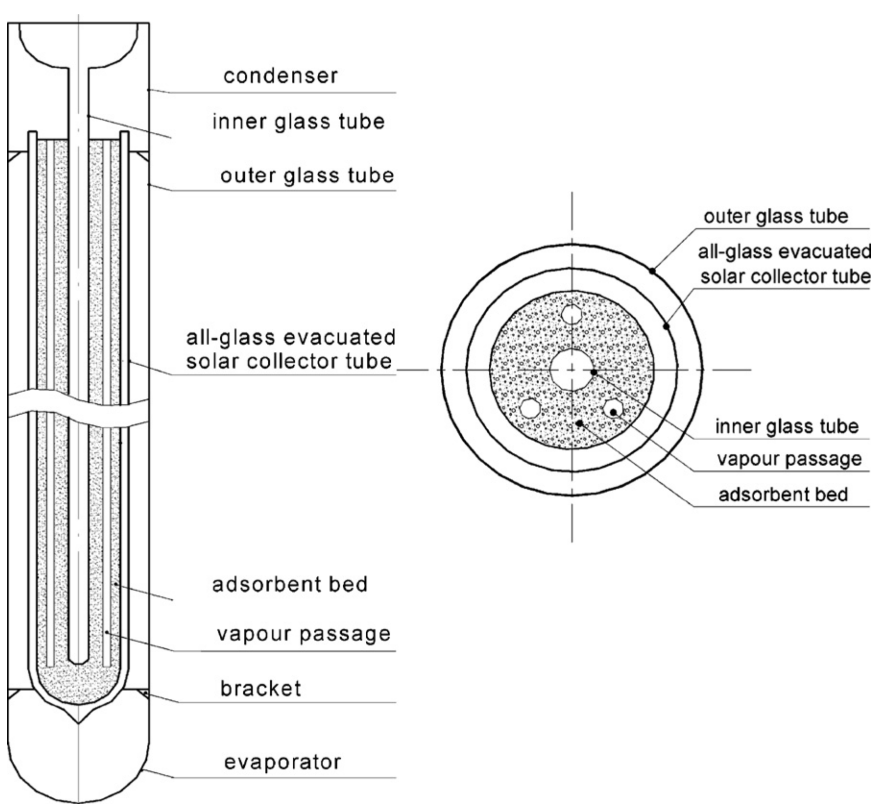

Fig. 6. Sketch of the solar cooling tube and its cross section [54].

that the highest adsorbent bed temperature is below 110 °C at solar radiation value between 15.3 and 17.1 MJ/m^2. The freeze proof solar cooling tube's cooling capacity was about 87–99 kJ, and the COP was more than 0.11 when the evaporation temperature was about −4 °C. Zhao et al. [49] developed a dynamic model based on coupled diffusion driving and temperature driving adsorption mechanism of the ART in which activated carbon-methanol was selected as the working pair for either refrigeration or air conditioning purposes.

9. Advanced systems

Refrigeration technologies that depend on the physisorption and operate in a closed cycles are variant. The typical physisorption basic cooling cycle has been subjected to many modifications and developments to increase and enhance the system performance. These advanced systems include many operation schemes like the continuous heat recovery cycle, mass recovery cycle, thermal wave cycle, convective thermal wave cycle, cascaded multi-effect cycle, and hybrid heating and cooling cycle.

9.1. Internal vapor mass recovery cycle

Internal vapor recovery or mass recovery process can play an important role to better the performance of adsorption refrigeration cycle [56]. The mass recovery process is usually used to increase the mass of the desorbed refrigerant by expanding the span between the minimum and maximum concentration. In this way, the refrigeration effect could be increased [57]. Coefficient of performance might be increased or decreased with mass recovery process due to different working conditions [56]. This cycle is very suitable for low generation temperatures especially those obtained by solar collectors. The schematic diagram of the internal vapor recovery system is demonstrated in Fig. 7. The system consists mainly of two adsorption beds that could be connected together through valve MR. The double bed adsorption cooling system is discussed previously in Section 4 and Fig. 1. It is obvious that, at the end

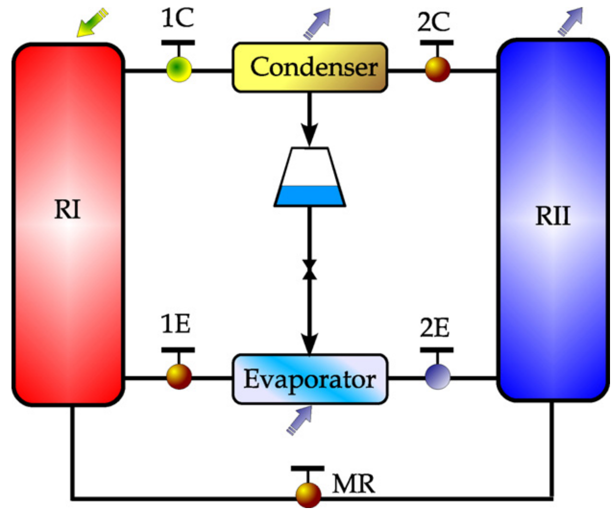

Fig. 7. Schematic diagram of the internal vapor recovery adsorption cooling system.

of each half cycle, one reactor is hot, RI, and at the high condenser pressure. Whereas the other reactor, RII, is cold and at the low evaporator pressure. During the next half cycle time, the bed RI should be cooled to reduce its pressure to the evaporator pressure. While the bed RII is to be heated, to raise its pressure to the condenser pressure. Obviously, a part of this pressurization–depressurization process can be achieved quickly through interconnecting the two reactors by opening valve MR until a pressure balance is reached between the beds. Since the vapor recovery process is rapid and only mass is transferred, it is considered an adiabatic process in which, the two bed pressure reaches an equilibrium pressure of $P_m = (P_{con} + P_{evap})/2$. However, the isothermal mass recovery process was studied by Wang et al. [58] and Taylan et al. [59].

The mass recovery cycle is demonstrated by the loop 1–1′–2′–3–3′–4′–1 on the Clapyron diagram in Fig. 8. It is noticed

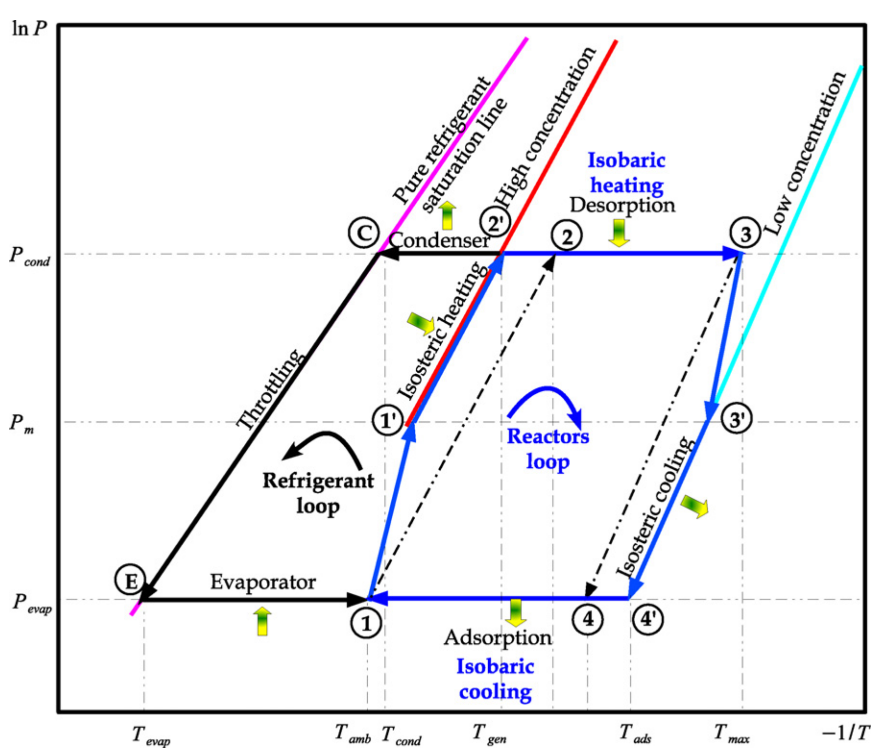

Fig. 8. Clapeyron diagram of the internal vapor recovery adsorption cooling system.

that the mass recovery process causes an increase in the adsorbed refrigerant mass during the adsorption process, from $\mathcal{X}_1$–$\mathcal{X}_4$ to $\mathcal{X}_1$–$\mathcal{X}'_4$. As a consequence, the refrigerant mass desorbed during the generation process is increased as well, from $\mathcal{X}_2$–$\mathcal{X}_3$ to $\mathcal{X}'_2$–$\mathcal{X}_3$. This increase leads to an increase in the cooling effect and therefore an increase in the cycle COP compared with the basic two-reactor cycle. In a comparison of COP between the two bed basic type cycle and the two bed mass recovery cycle, Wang [60] has found that the increased COP is in the range of 10—100% and the mass recovery cycle is practically suitable for low generation temperatures.

Akahira et al. [61] investigated the performance of two-bed, silica gel–water adsorption refrigeration cycle with mass recovery process. The results show that the cooling capacity of mass recovery cycle is superior to that of conventional cycle and the mass recovery process is more effective for low regenerating temperature. These results were confirmed by the same authors in another following experimental work [62]. A solar or waste heat driven three-bed adsorption cooling cycle employing mass recovery scheme was simulated and investigated by Khan et al. [63]. The chiller is driven by exploiting solar/waste heat of temperatures between 60 and 90 °C with a cooling source at 30 °C for air-conditioning purpose. The performance of the three-bed adsorption chiller with mass recovery scheme was compared with that of the three-bed chiller without mass recovery. It is found that cooling effect as well as solar/waste heat recovery efficiency of the chiller with mass recovery scheme is superior to those of three-bed chiller without mass recovery for heat source temperatures between 60 and 90 °C. However, COP of the proposed chiller is higher than that of the three-bed chiller without mass recovery, when heat source temperature is below 65 °C. The performance of an advanced three-bed adsorption chiller containing silica gel–water pair and with a mass recovery cycle has been experimentally investigated by Uyun et al. [64]. The temperature and pressure of various components of the chiller were monitored to observe the dynamic behavior of the chiller. The performances in terms of the coefficient of performance (COP) and specific cooling power (SCP) were compared with a conventional single stage. The results show that the proposed cycle produces COP and SCP values superior to those of the conventional single stage cycle for heat source temperature below 75 °C.

9.2. Heat recovery regeneration cycle

In fact, in the basic cycle, a substantial portion of the heat rejected from the bed being cooled is at a high temperature. Ideally, this allows for substantial regeneration of that rejected heat to reduce the thermal energy input required to heat the beds [65]. The heat recovery regeneration strategy is used to improve the adsorption cooling COP and the SCP. It usually operates with two beds that have a uniform temperature distribution along the solid adsorbent media. The sensible heat as well as heat of adsorption from the hot bed that is to be cooled is used to heat the other cold bed. By using more than two beds, more heat recovery and better COP can be obtained. However, the system will be complicated from point of view of the practical operation. Fig. 9 shows a schematic diagram for the simple two beds semi-continuous heat recovery system. A half operating cycle of the system compromises two consecutive stages. The first stage starts by cooling of adsorber RI and heating adsorber RII using the recovered heat coming from adsorber RI. In this stage the two circulating pumps drive the thermal fluid in the circuit between the two reactors while the connection to the heater and cooler are blocked. In this position, the three-way valve VI is opened in direction 1e and the other three-way valve is opened in direction 2e. The second stage starts when both of the reactors are nearly at the same temperature. The bed RI is connected to the cooler, valve VI has the position 1c, whereas reactor RII is connected to the heater, valve VII has the position 2h. This stage continues till

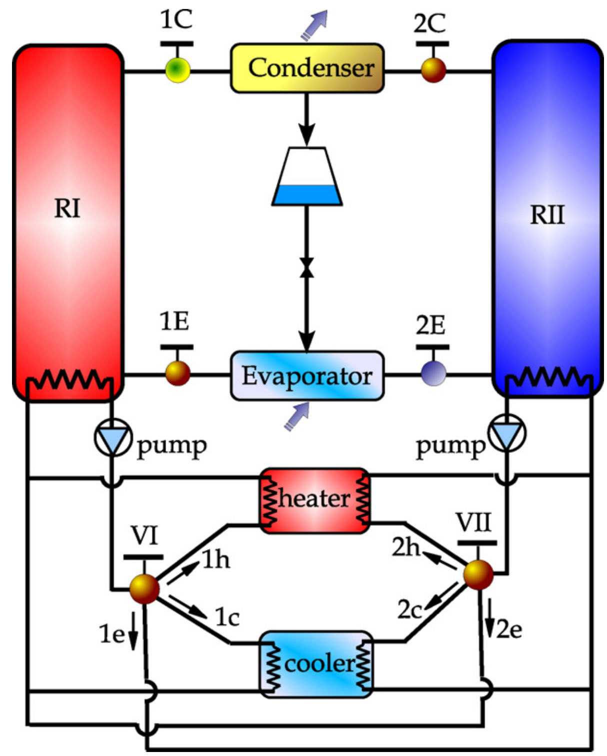

Fig. 9. Schematic diagram of the simple two beds heat recovery adsorption cooling system.

the end of the adsorption process in RI and desorption process in RII. The second half of the cycle is similar to the first half with the processes between adsorber RI and RII are exchanged. Depending on the time period in the heat recovery stage, the sensible heat or both the sensible followed by the adsorption heat can be recovered from the hot reactor to the cold one. However, if the adsorption temperature at the evaporator pressure is less than the generation temperature at the condenser pressure, the adsorption heat cannot be recovered.

The thermodynamic cycle of the simple two bed heat recovery system is drawn on Clapeyron diagram in Fig. 10. Where the first part of each half cycle is a sensible heat transformation process from the hot reactor, process 3–5, causing depressurization to the cold reactor, process 1–2, causing pressurization. Followed by a transfer of the latent heat of adsorption, process 4–6 and process 2–5. If the COP of the basic single effect cycle is Q_{evap}/Q_{des}, the COP of the cycle with heat recovery of Q_{reg} will be $Q_{evap}/(Q_{des} - Q_{reg})$. Calculations show that over 80% of the heat required for bed heating is available from the bed being cooled without violating the second law of thermodynamics [65]. The cycle COP can be increased more than 25% by heat recovery process [66].

Critoph [67,68] presented and simulated a continuous multiple-bed thermal regenerative rotary adsorption system for cooling applications. The system is based on a number of simple tubular adsorption modules circumferentially about a rotational axis partly within a toroidal conduit, Fig. 11. A single module is comprised of a generator and a receiver/condenser/evaporator, Fig. 12. A heat transfer fluid flows from an inlet of the conduit to the outlet in counter-flow with respect to the rotational movement of the adsorbent modules. Separate fluid channels encase the evaporation/condensation zones of the vessels to enable transfer of heat between the vessels and the fluid flowing in channels. A single generator consisting of a 12.7 mm stainless steel tube lined with 3 mm of monolithic active carbon has been manufactured. A complete module has been tested in a simple rig, which subjects

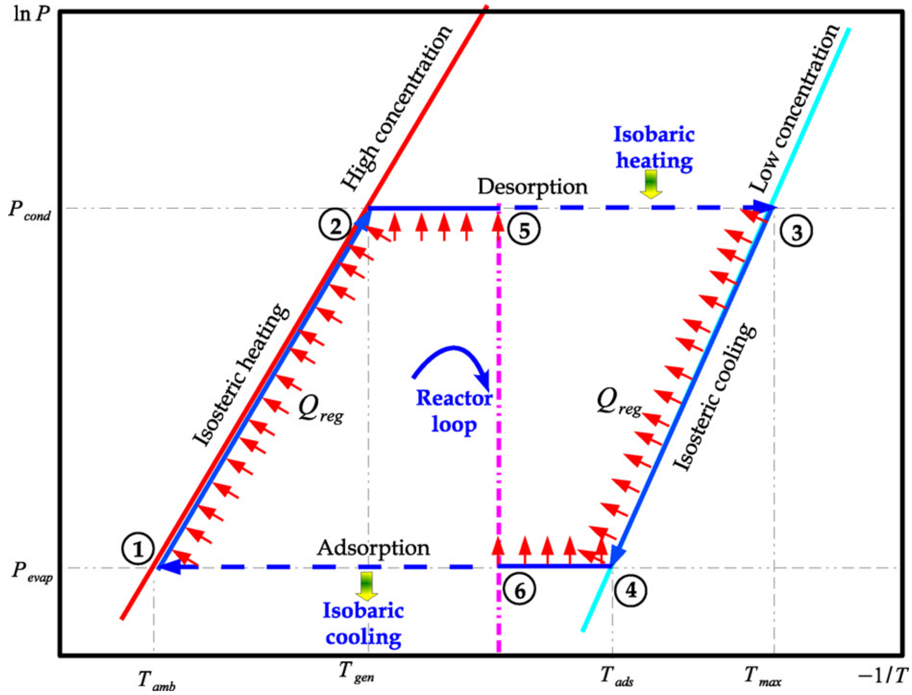

Fig. 10. Clapeyron diagram of the simple two beds heat recovery adsorption cooling system.

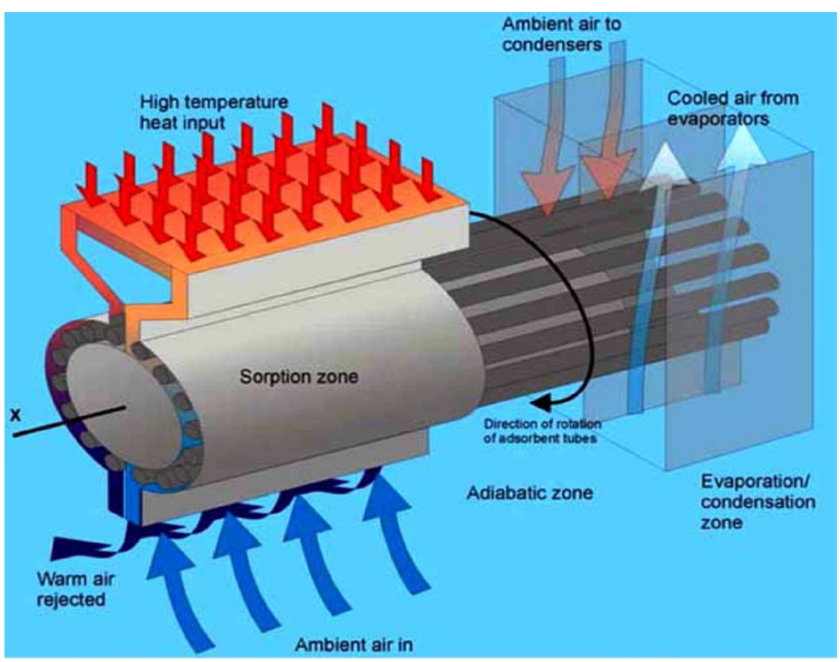

Fig. 11. Rotary thermal regenerative sorption device [69].

it to alternating hot and cold airstreams, desorbing and adsorbing ammonia. A complete system, consisting of 32 modules has been modeled in detail and its predicted performance is presented. Key parameters have been varied and their effect on the performance is discussed. As a result the compressive device is capable of achieving higher efficiencies than existing adsorption devices.

The experimental study for silica gel–water adsorption chillers with and without a passive heat recovery scheme is presented by

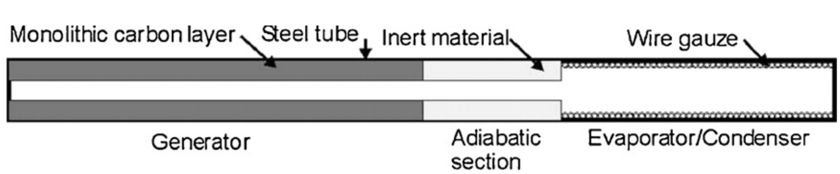

Fig. 12. The adsorption module [71].

Wang et al. [70]. Results showed that the passive heat recovery scheme improves the COPs of a two-bed chiller and a four-bed chiller by as much as 38 and 25%, respectively, without any effect on their cooling capacities. The highest COPs achieved with a two-bed and four-bed chillers are about 0.46 and 0.45, respectively. These are measured at a hot-water inlet temperature of 85 °C, cooling-water inlet temperature of 29.4 °C and chilled-water inlet temperature of 12.2 °C.

A predictive two-dimensional mathematical model of an adsorption cooling machine consisting of a double consolidated adsorbent bed with internal heat recovery is reported by Maggio et al. [72]. The results of a base-case, demonstrated that the COP of the double bed adsorption refrigeration cycle increases with respect to the single bed configuration. However, it was verified that, in order to maximize also the specific power of the machine, the adsorbent beds must have proper thermophysical properties. Wang and Chua [73] investigates the efficiency of two distinct heat-recovery schemes applied to the two-bed silica gel–water adsorption chiller. The purpose of these heat-recovery schemes is to improve the chiller performance and substantially improve the coefficient of performance. The two different schemes offer the same cooling capacity and similar COP boosting capability with the difference in COP by using the two schemes is less than 5%. For a real heat recovery adsorption system, the heat capacity of the metallic adsorber and also the thermal fluid will have a strong influence on the system COP. It is obvious that the COP decreases significantly if the ratio the total heat capacity ratio increases [60]. Good heat transfer should be considered in the real design of an adsorption system in order to shorten the cycle time and increase specific cooling power (SCP), which may need to increase heat transfer area by finned tubes. Wang [60] has designed and introduced an adsorber that uses activated carbon-methanol and used in air-conditioning applications, Fig. 13. The adsorber is a kind of a plate-finned tubes which is successful for both heat transfer and heat capacity control. The plate-finned tubes with a diameter of 9.5 mm are used for heat transfer between the thermal fluid and the adsorption bed, the aluminum plates are incorporated with the tubes to extend heat transfer surface. Adsorbent bed is packed by metallic net with the heat exchanger tubes, and a space to the shell is kept to allow gas flow channels for enhancing mass transfer. Inside the adsorber an extra mass flow channel is designed to separate the bed into two halves, [60]. Al Mers et al. [74] used a finned tube reactor and run an analysis of the sensitivity of the COP versus the geometrical parameters of the reactor, and the fins thickness and number.

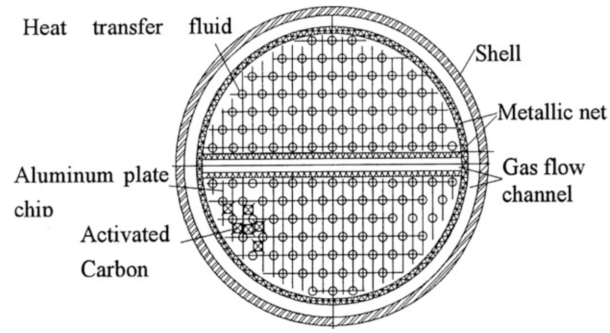

Fig. 13. Plate-finned tubes adsorber [60].

9.3. Mass and heat recovery cycle

Both the internal vapor mass recovery process and the regenerative heat recovery process, if combined together in the same cycle, a double effect enhancement of the cycle performance can be attained. Usually the mass recovery process takes place first followed by the heat recovery process. In fact, this merging strategy results in a significantly larger refrigeration effect and a better COP of the double effect mass and heat recovering system compared with either mass only or heat only recovery systems. The cycle time for mass and heat recovery will be much shorter and it will certainly enhance the cycle with higher cooling/heating power [56]. The principle operation and the thermodynamic cycle of the ideal mass and heat recovery system with two reactors is described in Clapeyron diagram as shown in Fig. 14. The first part of each half cycle is the mass recovery process, paths 1–1′ and 3–3′. Then the sensible heat recovery process proceeds with the heat transferred from the hot bed to the cold one, paths 1′–2′ and 3′–4′. Following is the adsorption latent heat recovery process, paths 2′–5 and 4′–6.

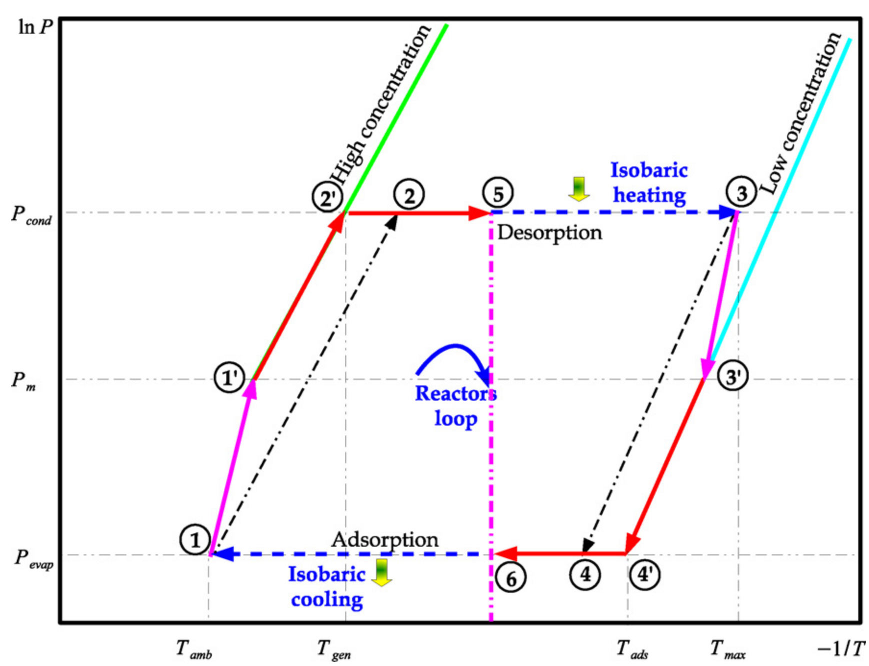

Fig. 14. Clapeyron diagram of a two bed double effect mass and heat recovery adsorption cooling system.

It has been shown that the best cycle performance is obtained when mass recovery is followed by sensible and adsorption heat recovery. For the activated carbon-methanol air-conditioning system, a COP over 0.6 can be achieved when the generation temperature is 80 °C. If the generation temperature reaches 120 °C, the COP will be close to 0.8 [66]. Furthermore, about 35% of the total energy transmitted to each reactor can be internally recovered, including part of the latent heat of sorption [75].

In their study on an adsorption air conditioner [56] found that the mass recovery process will promote the cycle COP or not, depending on the operating conditions. Moreover, the cycle with mass recovery and heat recovery has the highest COP among the calculated cycles. It is 30% and 10% higher than the basic cycle and heat recovery cycle, respectively. Wang et al. [58] introduced the definitions of the pressure coefficient of mass recovery to describe the degree of mass recovery and temperature coefficient of heat regeneration to describe the degree of heat regeneration. Besides, it is helpful to determine if and when sorption heat can be regenerated, as well as the combined effects of heat regeneration and mass recovery on adsorption refrigeration cycles.

A numerical transient model describing an adsorption refrigeration system based on the zeolite–water pair and incorporating a combined heat and mass recovery cycle is proposed by Leong and Liu [76]. The model describes the heat and mass transfer balance equations in two-dimensions in the adsorber in detail and is solved by control volume method. Internal and external mass transfer limitations which are neglected by many researchers are considered in the model since they have significant effects on the performance of the adsorption cooling cycle. The linear driving force equation is used to describe the micro mass transfer limitation in this model. This numerical model can be used in system optimization and design of adsorption cycles. The numerical results show that the mass recovery phase is very short (about 50 s) compared to the whole cycle time for the specified operating conditions. By using only the mass recovery cycle, the COP and SCP can be improved by about 6% and 7%, respectively, compared to the basic cycle. For the combined heat and mass recovery cycle with the given conditions, the calculated values of the COP and SCP are 0.651 and 27.58 W/kg, respectively. Although there is a significant increase in COP (by about 47%) compared to the basic cycle, there is an accompanied reduction in SCP by about 40%. The authors concluded that the selection of the operating conditions should consider the specific demands of the different systems. Similar results are obtained by Ng et al. [77] who performed experimental study on a four-bed adsorption chiller with heat and mass recovery schemes. They found that, when both the heat and the mass recovery schemes are employed at a rating point of maximum cooling capacity, the chiller COP could increase further to as much as 48%.

9.4. The thermal wave heat regeneration cycle

The thermal wave cycle is another attractive way for heat regeneration as in the heat recovery cycle. Thermal wave adsorption

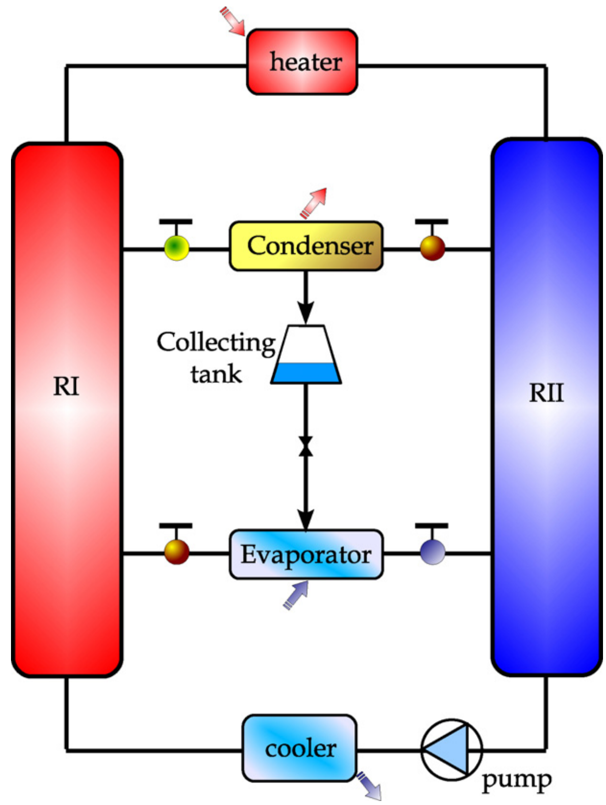

Fig. 15. Schematic diagram of a two bed thermal wave heat regeneration adsorption cooling system.

cooling cycles seems to be a promising technology to meet future air conditioning demand without introducing significant new electrical loads, especially in areas similar to the Mediterranean coast where summers are long and hot and cycles can be driven using solar energy [59]. This technique was first introduced and patented by Shelton [78]. The main objective of the thermal wave heat regeneration cycle is to minimize both the heat added to the system and the heat rejected to the ambient. Thereby, increasing the system thermal efficiency and the cold production capacity. In this cycle, a temperature gradient is established along the reactors lengthwise direction in order to establish a thermal wave in the bed moving axially thereof. The schematic representation of this concept is shown in Fig. 15 for a two bed system. It is noticed that this system is practically simpler than the previously discussed heat recovery technique. In the thermal wave heat regeneration system, a single heat transfer fluid such as high temperature oil circulation loop is passing through the two beds, the cooled, and the heater. The heater heats the heat transfer fluid to the upper operating temperature while the cooling heat exchanger cools the heat transfer fluid to the lower operating temperature. Moreover, each bed has a heat exchange arrangement that allows the heat transfer fluid to move

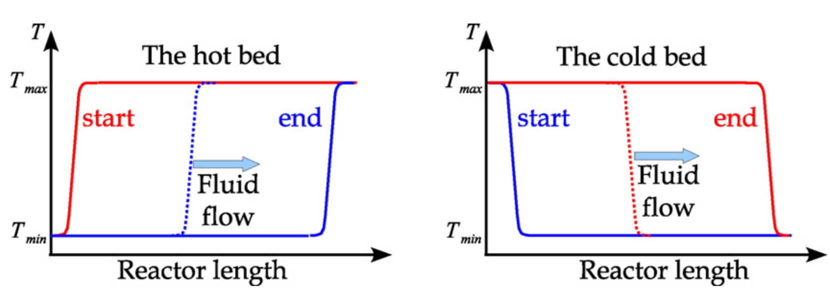

Fig. 16. The thermal wave temperature profile.

generally axially through the reactor in a single pass. The cooler is connected between one end of the bed heat exchangers while the heater is connected between the other ends of the bed heat exchangers. The circulating pump moves the heat transfer fluid in either direction around the circuit.

During operation, one of the solid adsorbent beds is being cooled while the other bed is being heated with the heating and cooling processes being reversed so that both beds are heated and then cooled during a cycle. As the hot heat transfer fluid out from the heater passes through the cold reactor, it is cooled to the bed temperature before it reaches the exit end of the bed so that the exiting heat transfer fluid is at the initial cool bed temperature, Fig. 16. As the thermal wave moves lengthwise of the bed, the temperature of the heat transfer fluid exiting the bed remains at the initial cold bed temperature until the thermal wave reaches the exit end of the bed. The cooler then cools down the already cooled heat transfer fluid to the lower operating temperature for the system. As the cold fluid passes through the hot reactor, it is heated to the bed temperature well before it reaches the exit end of the bed so that the exiting heat transfer fluid is at the initial hot bed temperature. As the thermal wave moves lengthwise of the bed, the temperature of the heat transfer fluid exiting the bed remains at the initial hot bed temperature until the thermal wave reaches the exit end of the bed. The heater then reheats the heat transfer fluid back to the upper operating temperature for recycling. When the thermal waves reach the exit ends of the beds, the flow of the heat transfer fluid is reversed so that the heated bed is cooled and the cooled bed is heated [78].

Using this method, a cooling COP on the order of 1.5–2.0 was achieved using water as the refrigerant and zeolite as the adsorbent [78]. Two factors mainly influence the characteristic of the thermal wave heat regeneration system. These are the fluid velocity and the equivalent heat transfer coefficient between the thermal fluid and the adsorbent. To achieve ideal thermal wave, the velocity of the fluid should be very low and the equivalent heat transfer coefficient should be high enough. However, low velocity of the heating fluid greatly restricts the specific cooling power of the system, and the increase of the equivalent heat transfer coefficient is greatly limited due to the characteristic of the porous adsorbent [66].

A thermodynamic analysis for a thermally driven ammonia/zeolite adsorption pair single stage heat pump with two beds and utilizes a simple heat transfer fluid circulating loop for heating and cooling two solid adsorbent beds is performed by Shelton et al. [65]. The heat transfer fluid loop also serves to transmit heat recovered from the adsorbing bed being cooled to the desorbing bed being heated. A square wave representation for the true shape of the thermal wave is used. This square wave is assumed to stop short of the bed ends to account for realistic finite wave forms. The square wave model is integrated into a thermodynamic cycle which provides detailed information on the performance of the beds as well as the COP and the heating and cooling outputs of the heat pump system. Significant cycle design and operating parameters are varied to determine their effect on cycle performance. The thermodynamic analysis of the cycle shows heating coefficients of performance of 1.87. Harkonen et al. [79] used a ramp wave model for the true shape of the thermal wave. The effect of the bed thickness on the waves is taken into account by assuming that the wave in the bed comes a little while after the wave in the fluid. Moreover, some examples for the zeolite/ammonia and zeolite/methanol pairs are discussed and the model is also compared with the existing old models by Shelton et al. [65] which overestimate the COP of the process. A research program aimed to develop a gas-driven heat pump in which the regeneration is accomplished by circulating a heat transfer fluid through two solid sorbent beds is reported by Miles et al. [80]. The development program uses thermal wave heat regeneration in which over 70% of the total bed input heat required per cycle is supplied by regeneration. In another study, Miles and Shelton [81] presented experimental results from a two-bed system which uses the activated carbon/ammonia pair. Many different bed designs and a number of cycle variations have been built and tested. They concluded that, through the use of a thermal wave regeneration concept, the efficiency of the system can be significantly increased. Using the experimental data a cooling seasonal COP of 0.76 and a heating seasonal COP of 1.21 were calculated.

In another study, Ben Amar et al. [82], an adsorptive heat pump system with the temperature wave heat generation is numerically analyzed, using a two-dimensional model which takes into account both heat and mass transfer processes inside the adsorber. In the model, the gaseous phase flow is described by Darcy's law. Numerical simulations are performed on two adsorption systems: zeolite NaX–water and activated carbon AX21–ammonia. The effects of the operating parameters, such as cycle time, permeability and heating temperature, on the cooling coefficient of performance and the power of cold production are discussed. Moreover, the temperature wave regenerative heat pump system is compared to the basic adsorptive heat pump process. The study showed that a COP greater than 1 and a power of cold production of near 200 W per kg of adsorbent could be obtained. Sun et al. [83] introduced a numerical analysis of an adsorptive heat pump system with thermal wave heat regeneration is presented by. They proposed a two-dimensional model taking into account axial heat transfer in the circulating fluid and radial heat conduction in the adsorbent bed. The time constants for heat exchange and heat conduction in the adsorbent bed are derived using the moment analysis and can be used to quantify the relative importance of the two heat transfer processes. The effects of the thermal conductivity and the cycle time on the process performance are also presented. They concluded that the performance of an adsorptive heat pump system using a traditional packed-bed would be too low, even with a heat regeneration, and therefore a significant enhancement of heat transfer properties inside the adsorber is necessary.

A local equilibrium model has been developed by Sward et al. [84] for a thermal-wave adsorptive refrigeration utilizing waste heat to assess wave shapes and study process variations. The model is utilized to examine the performance of adsorption refrigeration cycles powered by low temperature waste heat sources of 373–393 K. The predicted performance of the base cycle that utilizing a water/NaX zeolite adsorbate/adsorbent pair at 393 K heat source, 303 K condenser, and an evaporator temperature of 278 K gives a COP of 1.24. Models are presented by Taylan et al. [59] for ideal thermal wave adsorption cooling cycles without mass recovery, with adiabatic mass recovery and with isothermal mass recovery. Coefficient of performance (COP) values obtained from simulations is compared with the results of a reversible cycle and previously developed models for a simple cycle and heat recovery cycle with two spatially isothermal beds. The effects of maximum and minimum bed temperatures, bed's dead mass, and condensation and evaporation temperatures on COP were investigated as well. The authors concluded that the thermal wave cycle has significantly higher COPs than the simple and 2SIB cycles. For the conditions investigated, adding mass recovery to the thermal wave cycle does not affect its COP significantly. The COP of the thermal wave cycle increases with increasing maximum bed and evaporation temperatures and decreasing minimum bed and condensation temperatures. Unlike for the simple and 2SIB cycles, variations in the bed's dead mass have minimal impact on COP.

9.5. The convective thermal wave heat regeneration cycle

It is well known that heat transfer is a major problem in adsorption cycles due to the poor heat transfer in adsorbent beds which results in a long cycle times and a low power density [85]. This

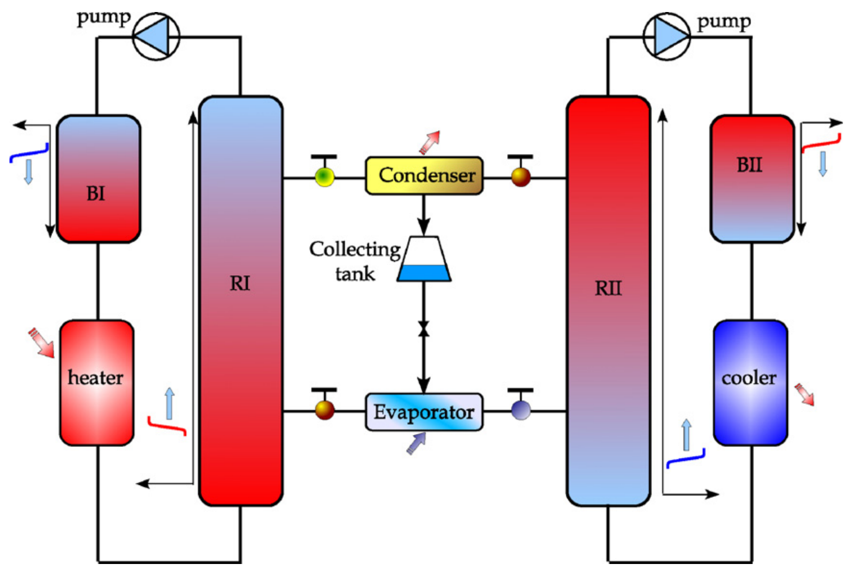

Fig. 17. The convective thermal wave adsorption system.

problem can me tackled in many different ways in which heat transfer can be improved. These include, but not restricted to, the use of extended surfaces within the adsorption bed as suggested by Wang [66] and is shown in Fig. 13. Another different approach suggested the use of the convective thermal wave technique. In 1992, Critoph [86] has introduced the convective thermal wave cycle which has the same concept as a of the basic thermal wave cycle but in a modified version. In this cycle, the thermal fluid is eliminated and the refrigerant itself is used to directly transfer heat from or to the adsorption reactors by direct contact. This can be done by heating or cooling the refrigerant before inlet to the reactor and circulate it through the adsorption media inside the bed [87]. Therefore, the refrigerant circulation process provides both a convective heat and mass transfer within the adsorption media. The effectiveness of the heat transfer process in this concept is mainly due to the large contact surface area between the porous media and the refrigerant. Therefore, the equilibrium state is reached rapidly and the cycle time is reduced. A practical schematic of the convective thermal wave system is shown in Fig. 17 This system makes use of four beds that are in effect heat exchangers of very compact and have high surface areas. Two beds are active and contain the adsorption media whereas the other two beds are inert and are filled with non-reactive particles such as steel balls to act as energy exchange media. The use of an inert bed with the convective wave cycle gives highly effective heat recovery and correspondingly good COPs [87]. During the first half of the cycle, active bed RI is heated and desorbs the refrigerant whereas the other active bed RII is cooled and adsorbing the refrigerant coming from the evaporator. In the heating loop of RI, the pump circulates the refrigerant vapor through the initially hot inert bed BI and the heater. When the refrigerant vapor pressure reaches to the condenser pressure, some of the desorbed vapor goes toward the condenser and the other recirculates in the heating loop again. In the cooling loop of RII, the pump circulates the refrigerant vapor through the initially cold inert bed BII and the cooler. When the refrigerant vapor pressure drops to evaporator pressure, some of the vapor coming from the evaporator is adsorbed and the other recirculates in the cooling loop again. When this half cycle is completed, the pumps are reversed and heating and cooling processes are exchanged to continue the second half cycle.

Critoph [85] modeled a forced convection heat pump cycle that uses the carbon–ammonia pair. The cycle appeared to offer good power densities of 1–3 kW/kg of adsorbent in the range of cases modeled. Heat pumping COPs of about 1.3 are predicted for the situation modeled, but the regenerative nature of the cycle gives rise to the belief that COPs up to a maximum of 1.9 may be possible with optimized energy management within the system. In another study, Critoph [87] presented a thermodynamic modelling, based on measured heat transfer and porosity data for the forced convection adsorption cycle. The system makes use of an inert bed that gives highly effective heat recovery and correspondingly good COP. Initial simulations of the cycle imply that with careful choice of carbon, grain size, bed size and flow rates, it will be possible to obtain a satisfactory combination of COP and power density. The author predicts a cycle COP (for a specific carbon) of 0.95 when evaporating at 0 °C and condensing at 42 °C. In a similar following paper by Critoph [88], a predicted COP of 0.90 when evaporating at 5 °C and condensing at 40 °C, with a generating temperature of 200 °C and a modest system regenerator effectiveness of 0.8 were obtained. Moreover, the experimental heat transfer measurements and cycle simulations show the potential of the concept to provide the basis of a gas-fired air conditioner in the range 10–100 kW cooling.

A simulation example for the convective thermal wave generation cycle using activated carbon fiber–ammonia pair has been discussed by Wang [66]. The author reported that the refrigeration COP is 0.78 for a 1 kg ACF in the adsorber with 250 mm in length, condensing temperature of ammonia is 35 °C, heat driven temperature of 165 °C, cooling temperature of 40 °C, and evaporation temperature is −8 °C. A theoretical investigation of a convective thermal wave adsorption chiller was tackled numerically and parametrically by Tierney [89] with activated carbon-methanol working pair. The coefficient of performance (COP) is found insensitive to the heat capacity of the refrigerant vapor, or the effective thermal conductivity of the refrigerant. Moreover, increasing either the bed effective heat capacity or its effective axial conductivity strongly impeded performance. The author also found that good COPs and cooling powers were apparent if the parasitic pumping power was neglected. Realistic pumping power was possible only at the expense of relatively large machines and poor COP; the two attributes that the convective thermal wave machines are intended to enhance. The results discouraged the building of costly experiments. A different convective thermal wave cycle configuration which is called periodic reversal forced convective cycle is suggested by Lai [90]. The proposed cycle is based on the combination of convective thermal wave cycle proposed by Critoph [85] and periodic flow reversal cycles introduced by Lai and Li [91]. In

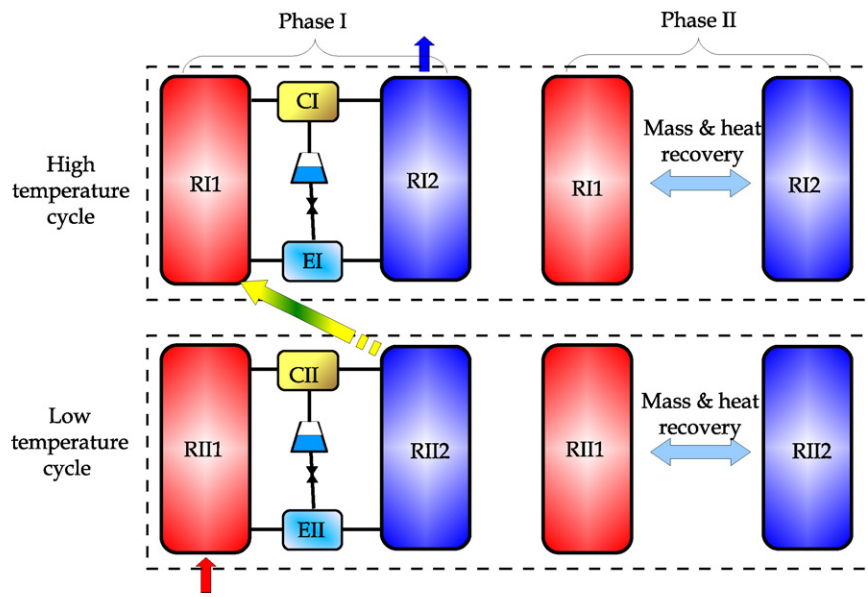

Fig. 18. Illustrative diagram for a two cycle cascaded system.

this cycle, improvement in performances is envisaged by applying the fixed bed adsorbers with the forced unsteady-state operation of periodic flow reversal to convective thermal wave cycle. The results of calculation for the zeolite 13X–water pair show that more than 0.9 of cooling COP and 125 W/kg adsorbent of specific cooling power (SCP) within this system could be possible.

9.6. The cascading systems

In case of high temperature heat source availability (over 150 °C), it is not practical to use a single cycle and in this situation the cascading arrangement of cycles is useful and provides high COP. In a cascading system, two or more different cycles operate in series at different temperature levels and with different working pairs. The cycle that works at high temperature level is the top cycle, whereas the cycle which works at low temperature level is the bottom cycle. In this way, the system consists of a multi-adsorption beds working with different working pairs, and therefore more than one condenser and evaporator may be used. The high temperature heat source is used to drive the high temperature cycle. In this way, the sensible heat and heat of adsorption at high temperature cycles is used to activate the desorption process at the lower temperature stages. Therefore, the entropy production due to the coupling between the external heat and the reactors is much less in a cascading arrangement than it is in a single cycle [92].

Due to water freezing, zeolite–water pair can operate only at evaporating temperatures higher than 0 °C. Besides that zeolite has a higher adsorption heat than other adsorbers, such as silica or active carbon. Moreover, this pair is able to operate cycles with large evaporating temperature lifts (70 °C or even more) where the evaporating temperature lift is the difference between adsorbing temperature and evaporating temperature [93]. Therefore, high temperature top cycle is usually operated by the zeolite–water pairs with the driving temperature ranging from 100 to 200 °C. Whereas, active carbon-methanol is limited to regenerating temperatures of the order of 150 °C due to methanol instability. That is besides, active carbon-methanol pair is well adapted to operate cycles with small evaporating temperature lifts (up to 40 °C). As a consequence, activated carbon-methanol or silica gel–water are normally employed for the low temperature stages with a generation temperatures below 100 °C. In a cascaded system, the heat regeneration and the internal vapor mass recovery or both can be introduced for the stages to better management of the available energy.

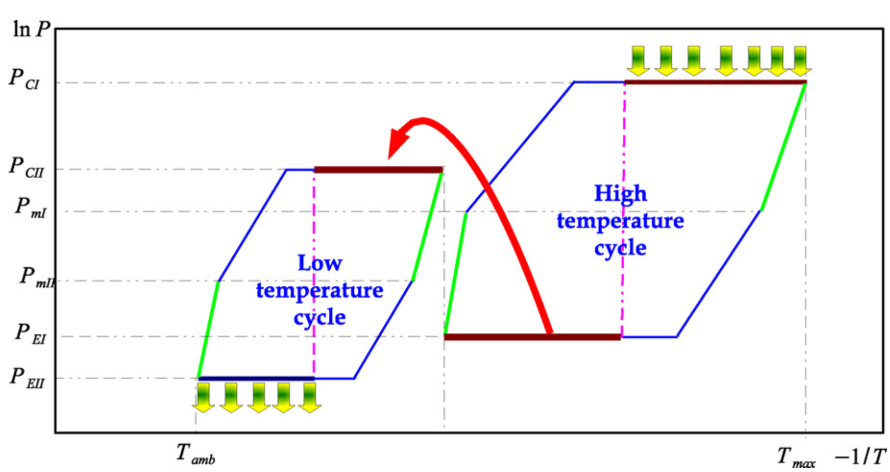

Fig. 19. Clapeyron diagram for a two cycle cascaded system.

The system configuration of a two cascaded adsorption refrigeration cycles operating in series is shown in Fig. 18. The first cycle is the high temperature cycle which consists of two reactors RI1 and RI2, a condenser CI, and evaporator EI. The second cycle is the low temperature cycle that consists of reactors RII1 and RII2, condenser CII, and evaporator EII. Both of the high and low temperature cycles are working with a double effect mass and heat recovery strategy. The working principle of the system is described with the help of Clapeyron diagram in Fig. 19. A half cycle operation of the system takes place in two phases. In the first phase, RI1 is heated by heat input and desorbs to the condenser CI while RI2 adsorbs the vapor from evaporator EI. The reactor RII1 is heated by the sensible and adsorption heat from RI1 and desorbs to EII while RII2 adsorbes the refrigerant vapor coming from EII and reject the adsorption heat to the ambient. In the second phase, a mass recovery followed by a heat recovery process takes place for both of the high and low temperature cycles reactors. The second half cycle is similar but with exchanging operations between each cycle reactors.

Most of the work done regarding the possibilities of cascading cycles is of theoretical studies whereas few experimental activities are found. The theoretical efficiencies of a solid adsorption heat pump are discussed as a function of the number of cascades by Meunier [92]. It was shown that for an infinite number of adsorbers with ideal heat recovery, the maximum achievable cooling COP equals 1.85 which corresponds to 68% of ideal Carnot COP. Meunier [93] proposed a cascading cycle, in which a single effect active carbon-methanol cycle is boosted by a double effect zeolite–water cycle. Results show that the COP of the active carbon-methanol cycle and zeolite–water cycle was 0.54 and 0.41, respectively, and with a total system COP of 1.15. Moreover, the one third of the cooling effect provides from the zeolite–water pair when the two other thirds provide from the active carbon-methanol pair.

Experimental work on a cascading adsorptive heat pump are reported by Douss and Meunier [94]. The cascading cycle consists of a two adsorber zeolite–water as a high temperature stage and an intermittent active carbon-methanol as a low temperature stage. Driving heat is supplied by a boiler to zeolite adsorbers while active carbon adsorber is heated by heat recovered from zeolite adsorber under adsorption. Evaporators from both basic cycles operate at the same temperature and contribute to the evaporating load. Experimental cooling COP is found to be 1.06, much more than the COP of an intermittent cycle (0.5) and more than the COP of a two adsorber zeolite water cycle (0.75). Wang [66] carried a study on a four-bed cascaded adsorption refrigeration system using the same working pairs. Two arrangements were studied. The first is a double effect system in which sensible and adsorption heat recovery are used to generate the low temperature stage. The second is a triple effect system in which the heat recovery of the desorbed vapor in the high temperature stage in addition to the heat used in a double effect arrangement are used to drive the low temperature stage. The triple effect system provides the maximum energy recovery from the high temperature stage. The analysis results in the relation between the cascading cycle COP and two independent stage cycle COP1 and COP2 as $COP = COP1 + COP2 + COP1 \cdot COP2$. A typical example is that if both stages have $COP = 0.6$, then the triple effect cascading $COP = 1.56$.

A two cycle cascaded silica gel–water adsorption refrigeration system that consists of four-bed mass recovery technique and is driven by low temperature heat source is discussed by Alam et al. [95]. The heat source rejected by one cycle is used to power the second cycle. Due to the cascading use of heat and cooling source, all major components of the system maintain different pressure levels. The proposed cycle utilize those pressure levels to enhance the refrigeration mass circulation that leads the system to perform better performances. It is seen that the cooling effect as well as COP of the proposed cycle is superior to those of the basic cycle. The performances of the cycle are compared with those of the two-stage cycle. Results also show that though the cooling effect of the proposed cycle is lower than that of two-stage cycle for heat source temperature less than 70 °C, the COP of the cycle, however, is superior to that of two-stage cycle for heat source temperature greater than 60 °C. In their study, Akahira et al. [96] investigated the performance of a four-bed, silica gel–water mass recovery adsorption refrigeration cycle with energy cascading and different pressures in adsorber and desorber. The using of the pressure deference of the adsorber/desorber heat exchangers helps to accelerate adsorption/desorption process. The proposed cycle was compared with the single-stage cycle in terms of SCP and COP. The results show that SCP of proposed cycle with cascading chilled water is superior to that of conventional, single-stage cycle and the proposed cycle has high advantage at low heat source temperature. The COP values of proposed cycle are also higher than those of single-stage cycle if heat source temperature is lower than 70 °C.

Liu and Leong [97] studied a cascading adsorption cooling cycle for refrigeration purposes with a three adsorbers. This cycle consists of two zeolite–water adsorbent beds and a silica gel–water adsorbent bed as the low temperature stage. Both heat and mass recovery are carried out between the two zeolite adsorbent beds and heat is also exchanged between the zeolite adsorbent and the silica gel adsorbent beds. This cycle is simpler than that proposed by Douss and Meunier [94], since it uses only one type of refrigerant (water) and requires only one condenser and one evaporator. The COP for the base case is found to be 1.35, which is much higher than the COP of an intermittent cycle (about 0.5) and a two-bed combined heat and mass recovery cycle (about 0.8). However, its specific cooling power (SCP) of 42.7 W/kg is much lower than that of the intermittent cycle. The performance of a cascading adsorption arrangement that utilizes the silica gel–water pair was studied by Uyun et al. [98]. The top cycle is a single stage cycle that is driven by an external heat source with a temperature between 90 and 130 °C. While the bottom cycle works in a mass recovery strategy and is driven by waste heat of sensible and adsorption heat of the high temperature cycle. The performances, in terms of the coefficient of performance (COP) and the specific cooling power (SCP), are compared with conventional cascading without mass recovery and single-stage cycles. The results show that in the comparison between the conventional and the proposed cycle at the observed heat source temperature, the COP and SCP values of the proposed cycle are superior to those of the conventional cycle. Additionally, the cascading adsorption cycles produce higher COP values but lower SCP values, in comparison with the conventional single-stage cycle. In another study, Marlinda et al. [99] have introduced a numerical investigation for a cascading adsorption system that utilizes condensation heat produced in the top cycle to drive the lower temperature cycle. Their proposed system uses silica gel/water as the working pair for both the high temperature and the low temperature cycles and consists of four adsorption reactors, a condenser and an evaporator. The performance of the proposed system is determined from dynamic simulations, and the results are compared with conventional single-stage cycle to which the chilled outlet temperature is equivalent. The results show that the double-effect cycle produces a higher coefficient of performance (COP) as compared to that of the conventional single-stage cycle for driving temperatures between 100 °C and 150 °C in which the average cycle chilled water temperature is fixed at 9 °C. Moreover, the COP of the double-effect cycle is more than twice that of the single-stage cycle when the temperature reaches 130 °C.

9.7. The multi-stage systems

Most of the cycles mentioned previously discussed require either high or medium temperature heat sources to operate. As

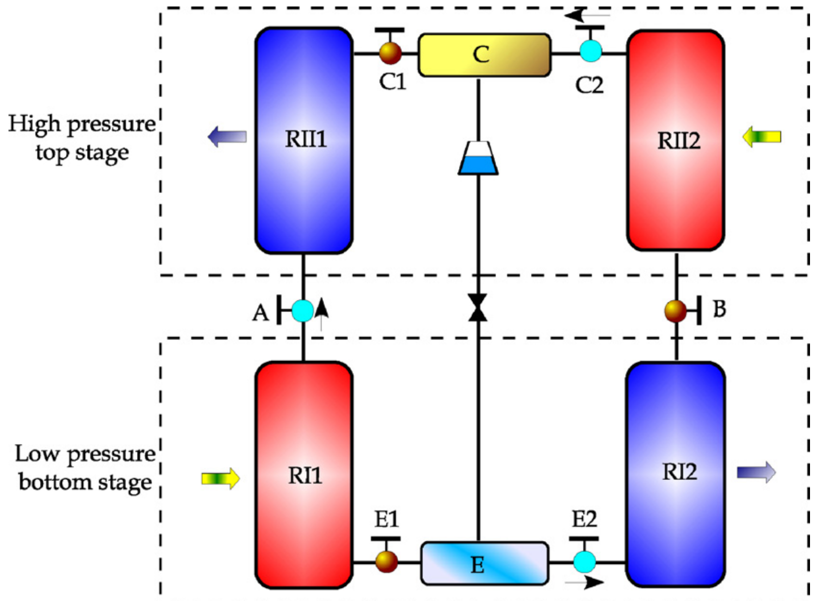

Fig. 20. Illustrative diagram for a two stage adsorption cooling system.

discussed by Alam et al. [100], a conventional silica gel–water adsorption cycle cannot be operational with the driving heat source temperature of 50 °C if the heat sink is at 30 °C or more. For those heat sources with relatively low temperatures or near-ambient temperatures (between 50 and 75 °C), the cycle multi-staging will be a good choice to operate the adsorption cooling system. However, the performances of multi-stage systems are lower than those operating with the basic cycles. Another drawback of multi-stage adsorption system is its fluctuation in delivered chilled water temperature [24]. The fundamental principle of a multi-stage cycle is to perform the desorption–condensation processes at higher pressure levels and by using the same working pair in a quasi-continuous operation. The refrigerant pressure rises during consecutive steps from evaporation to condensation level. The simplest type of the multi-stage adsorption system is the basic two stage cycle that was introduced by Saha et al. [23]. The two-stage cycle consists of four reactors, a condenser, and an evaporator. A schematic representation for this system is illustrated in Fig. 20 and the thermodynamic cycle on Clapeyron diagram is shown in Fig. 21.

A complete operating cycle is divided into two parts with two phases in each part. The first phase of the first half cycle includes a pre-heating process for the low and high pressure reactors, RI1(1–2) and RII2(5–6), respectively, and a pre-cooling process for the other two reactors, RI2(3–4) and RII1(7–8). The second phase of the first half cycle starts when the pressures in RI1 and RII1 reach the same value at the intermediate pressure P_{int} and when the pressures in RI2 and RII2 reach the evaporation and condensation pressures, respectively. At this time, valves A, C2, and E2 are

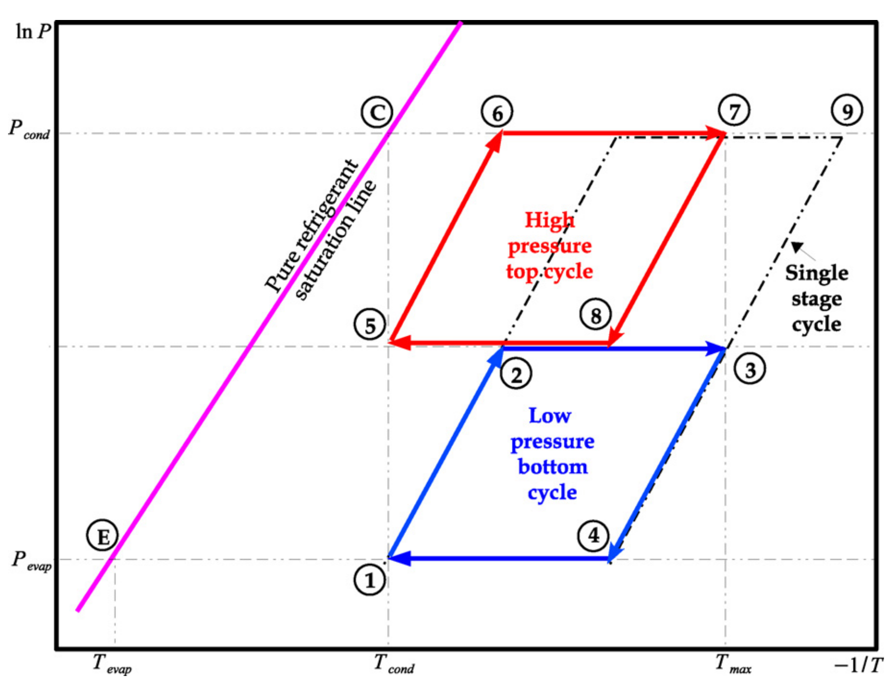

Fig. 21. Clapeyron diagram for the conventional and the basic two stage adsorption cycles.

opened whereas valves B, C1, and E1 are closed, Fig. 20. Refrigerant vapor desorbed from RI1, process 2–3, and flows toward the top stage high-pressure reactor RII1 where it is adsorbed, process 8–5. Cooling production takes place in the evaporator and the refrigerant vapor desorbed in RI2, process 4–1, whereas the high-pressure desorber RII2 release its refrigerant content into the condenser, process 6–7. The condensed refrigerant comes back to the evaporator via the expansion valve connecting the condenser and the evaporator. The second half of the cycle has the same operation but with the heating, cooling, and the valves operation modes are exchanged.

It is noticed from Fig. 21 that the multi-stage technique is able to use lower temperature heat sources compared to the basic conventional cycle, $T_7 < T_9$. That is besides the reduced regeneration temperature lift, $\Delta T_{reg} = T_6 - T_{cond}$ as a result of dividing the evaporating temperature lift, $\Delta T_{evap} = T_{cond} - T_{evap}$, into two smaller lifts. Therefore, the refrigerant vapor pressure rises via successive stages from evaporation to condensation level.

Some researchers investigated the performance of multi-stage adsorption refrigeration cycles. To exploit solar/waste heat of temperatures below 70 °C, a two stage chiller prototype using silica gel–water as the adsorbent refrigerant pair was designed and experimented in a study by Saha et al. [23] at Tokyo University of Agriculture and Technology. Experimental temperature profiles of heat transfer fluid inlets and outlets are measured. Results showed that the two-stage cycle can be operated effectively with 55 °C solar/waste heat in combination with a 30 °C coolant temperature with a COP of the two-stage chiller of 0.36 and a rated cooling capacity of 3.54 kW. The authors also concluded that flat plate solar collectors in any tropical climate can effectively produce the required driving source energy of the chiller making it superior to any other commercially existing cooling technology.

A two-stage adsorption chiller with silica gel as adsorbent and water as refrigerant was studied analytically to investigate the effect of operating and design parameters on the cooling capacity and COP of the chiller [100]. The cycle simulation was verified with the experimental data. Simulation results show that, for the base conditions, the two-stage adsorption chiller with 12.5 kg silica gel shows the highest COP, while cooling capacity and specific cooling capacity are proportional and inversely proportional to the weight of silica gel, respectively. Moreover, both the COP and the cooling capacity improve with the increase of overall heat transfer coefficient. The optimum value of the heat transfer coefficient for the base run condition is 1.5 kW/m^2 K.

In a paper by Khan et al. [101], a two stage chiller similar to that of Alam et al. [100] but with reheat and is driven by waste heat temperatures between 50 and 70 °C with a cooling source at 30 °C for air conditioning purposes is introduced. An analytic investigation of the adsorption refrigeration chiller is performed to determine the influence of the overall thermal conductance of sorption elements and evaporator as well as the adsorbent mass on the chiller performance. The analysis shows that the cycle performance is strongly influenced by the overall thermal conductance of sorption elements due to the sensible heating and cooling requirements resulting from batched cycle operation. The model is somewhat sensitive to the overall thermal conductance of the evaporator. A similar study to improve system performance was introduced by Khan et al. [102]. An analytic investigation on a re-heat two-stage chiller is performed to clarify the effect of thermal capacitance ratio of the adsorbent and inert material of sorption element, overall thermal conductance ratio of sorption element and evaporator along with silica gel mass on the chiller performance. Results show that cycle performance is strongly influenced by the sorption elements overall thermal conductance values due to their severe sensible heating and cooling requirements resulting from batched cycle operation. The effect of thermal capacitance ratio becomes significant with relatively higher mass of silica gel. It is also found that the chiller performance increases significantly in the range of silica gel mass from 4 to 20 kg.

The performance of an adsorption chiller with two-stage reheat cycle has been investigated experimentally in the study reported by Alam et al. [103].The performances in terms of specific cooling power (SCP) and COP are compared with those of conventional single and two-stage chiller. Results show that the reheat two-stage chiller provides more SCP values than those provided by conventional single-stage chiller while it provides better COP values for relatively low heat source temperature. The reheat two-stage chiller also provides almost same cooling capacity comparing with two-stage chiller for the low temperature heat source, while it provides higher COP value than that provided by two-stage chiller. Experimental results also show that the overall performance of the reheat two-stage chiller is always higher than that of conventional single and two-stage adsorption cycle even the temperature of the heat source is fluctuated between 55 and 80 °C. Oliveira et al. [104] presented the experimental results of an adsorption ice maker operating in double stage without mass recovery and under two kinds of mass recovery cycles. Two generation temperatures used in the experiments, 85 and 115 °C. At 85 °C, the cycle with mass recovery and two stage produced the highest cycled mass, rate of ice production, cooling capacity and its COP was similar to that obtained in the cycle with conventional mass recovery. That is besides a 42% more refrigerant mass generation than the cycle without mass recovery. Whereas, at generation temperature of 115 °C, the cycle with conventional mass recovery generated 37% more mass than the other two stage cycle. The best performance was obtained at generation temperature of 115 °C and with the cycle that employed conventional mass recovery.

The performance of a two-stage silica gel/water adsorption chiller with different mass allocations between upper and bottom beds has been investigated numerically [105]. It is found that the chiller can be driven effectively by the waste heat of temperature 55 °C with the heat sink at environment temperature. It is found that the cooling capacity of a two-stage adsorption chiller can be improved by allocating adsorbent mass between upper and bottom cycle. The improvement in COP values, however, is less significant. It is also seen that the two-stage chiller with 1:2 (upper/bottom) provides more cooling capacity than that provided by single-stage chiller if the heat source temperature is 80 °C. It is also seen that the improvement in cooling capacity is more significant for the relatively higher heat source temperature. Moreover, the cooling capacity can be improved up to 20% if the heat source temperature is 80 °C and the average outlet temperature is fixed at 7 °C and at mass ratio 2:5. In a similar study, a silica gel/water two-stage and four bed adsorption chiller with different mass allocations between upper and lower beds employing a reheat scheme was introduced [106]. The chiller is powered by waste heat or renewable energy source of temperature between 50 °C and 90 °C with a coolant of inlet temperature 30 °C for air conditioning purpose. The performance of the chiller employing the reheat scheme is compared with that of the two-stage conventional chiller with the reheat scheme using equal mass allocation. Results show that the cooling capacity can be improved up to 10.78% while the cooling water temperature is at 20 °C.

A thermally driven, advanced three-stage adsorption chiller utilizing low-grade waste heat of 50 °C and lower temperatures as the driving heat source, in combination with a heat sink of 30 °C is reported by Saha et al. [107]. The chiller consists of six adsorption beds that utilize the silica gel–water working pair and the system is designed for use in air conditioning. The performance of the chiller and the influence of operating conditions (temperatures, flow rates and adsorption–desorption cycle times) on cooling output, and chiller COP were investigated analytically. Simulation results showed that the three-stage chiller can be operated with

heat sources of 50 and 40 °C in combination with cooling sources of 39 and 30 °C, respectively. It was also shown that, for the chiller to operate effectively, heat sources of 50 °C require cooling sources between 35 and 20 °C, while heat sources of 40 °C need cooling sources in the range of 28–20 °C.

A more advanced three-stage adsorption system was proposed in another work by Saha et al. [108,109]. In these studied, a dual-mode silica gel–water adsorption chiller that utilizes effectively low-temperature solar or waste heat sources of temperature between 40 and 95 °C was investigated and simulated. Two operation modes are possible for the advanced chiller. The first operation mode is to work as a high efficient conventional chiller where the driving source temperature is between 60 and 95 °C. The second operation mode is to work as an advanced, three-stage adsorption chiller where the available driving source temperature will be lower than 60 °C (i.e., between 40 and 60 °C). The main innovative feature of the dual-mode chiller is the ability to operate with not only at small regenerating temperature lift (10 K) but also at mid regenerating temperature lift (50 K) and it is therefore attractive as an energy saver. Simulation results showed a poor efficiency in terms of cooling capacity and COP of the advanced three-stage mode. Moreover, the optimum COP values are obtained at driving source temperatures between 50 and 55 °C in three-stage mode, and between 80 and 85 °C in single-stage, multi-bed mode. A three stage six-bed adsorption chiller employing re-heat scheme and working with silica gel/water pair was introduced by Khan et al. [110]. The chiller is powered by waste heat or renewable energy sources of temperature between 50 and 70 °C along with a coolant of inlet temperature at 30 °C for air conditioning purpose. The performance of the six-bed adsorption chiller using re-heat scheme is compared with that of the six-bed chiller without re-heat. It is found that both the cooling capacity and the coefficient of performance of the three-stage chiller with re-heat scheme are superior than those of the three-stage chiller without reheat scheme.

9.8. The multi-bed systems

The multi-bed regenerative strategy is an extension from the two-bed operation. In a conventional two-bed adsorption chiller, the inherent restriction in the number of beds resulted in significant temperature fluctuation in all the components. On top of the fluctuation of the chilled water temperature, the peak temperature of the condenser outlet temperature that follows shortly after bed switching adds on to the instantaneous load of the cooling tower. The multi-bed scheme will serve to significantly slash the peak temperatures in both the evaporator and the condenser [111].

Through the use of a multi-bed regenerative strategy, Chua et al. [111] investigated the possibility of improving the recovery efficiency of waste heat to useful cooling by maximizing the cooling capacity and damping the chilled water outlet temperature fluctuation. A four-bed regenerative scheme is proposed and its performance was solved numerically. For the same waste heat source flow rate and inlet temperature, a four-bed chiller generates 70% more cooling capacity than a typical two-bed chiller. A six-bed chiller in turn generates 40% more than that of a four-bed chiller. The drawbacks are the requirement of two more sorption elements, which results in higher initial cost.

Saha et al. [24] reported a three-bed non-regenerative silica gel–water adsorption chiller. The three-bed chiller is able to work as high efficient single-stage adsorption chiller where driving source temperature is between 60 and 95 °C along with a coolant at 30 °C. The proposed three-bed strategy is an extension from the conventional two-bed operation. At any time, two adsorber beds will be in operation. The cycle simulation calculation indicates that the COP value of the three-bed chiller is 0.38 with a driving source temperature at 80 °C in combination with coolant inlet and chilled water inlet temperatures at 30 and 14 °C, respectively. The delivered chilled water temperature is about 6 °C with this operation condition. Simulation results also show that from the two to three beds, waste heat recovery efficiency is boosted by about 35%.

Numerical analysis of an adsorption cycle employing advanced three-bed mass recovery cycle with and without heat recovery is introduced [112]. The cycle consists of three silica gel adsorbent beds that can be divided into two cycles with different desorption mechanisms. The performances of three-bed, single stage, and mass recovery adsorption cycles are compared in terms of coefficient of performance and specific cooling power. The results show that by applying heat recovery to the cycle, better COP values will be produced compared to that without heat recovery. The results also show that there is an optimum point of adsorber mass distribution and desorption time that produces optimum performance.

The performance evaluation of a waste heat driven dual evaporator type three-bed adsorption cycle for cooling application is presented in Miyazaki et al. [113]. The adsorption chiller has two evaporators, three adsorbent beds, and a condenser, and the evaporators work at different pressure levels. The effects of several parameters on the specific cooling capacity and coefficient of performance were predicted by simulation. For the same operating condition, the SCC and COP of the dual evaporator type three-bed adsorption chiller were found to be 1.5 and 1.7 times higher than those of the two-bed single-stage adsorption chiller, respectively.

In a conventional adsorption chiller operation, the appropriate pre-heating/pre-cooling process enhances the cooling capacity, whereas the excess pre-heating/pre-cooling time abates the average cooling capacity. The cycle time allocation not only improves the cooling capacity and coefficient of performance, but also contributes in the reduction of delivered chilled water fluctuations. Miyazaki et al. [114] worked to enhance the performance of silica gel–water based adsorption chillers by cycle time allocation. The cycle time allocation allows the continuous cooling effect over the cycle without sacrificing the effect of pre-heating/pre-cooling. Simulation results showed that the new cycle time was effective for both RD type silica gel–water and $CaCl_2$-in-silica gel–water pairs, and the cooling capacity was increased as much as by 6%.

10. Conclusion

Traditional vapor compression machines are dominating electricity consumers and their refrigerants have high global warming as well as ozone layer depletion potentials. Utilization of a green energy type like solar energy is an option to solve the electricity and pollution problems. Solar powered adsorption refrigeration systems represent a promising substitute for those traditional vapor compression machines. They are proven to be suitable and applicable for refrigeration as well as air-conditioning applications. This kind of technologies is divided into physisorption and chemisorption systems. The physisorption machines include open and closed cycle operation. In this paper we presented a review on previous studies and developments of the solar driven closed cycle physisorption refrigeration systems. The discussion includes, experimental and numerical simulation studies as well as methods that are suggested to improve the system performance.

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